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NOVEMBER ★ 1942

INDUSTRIAL
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ge 679: Journal Section. American Society of Heating and Ventilating Engineers

Information on removal of refrigeration machines for use in war production is published this month

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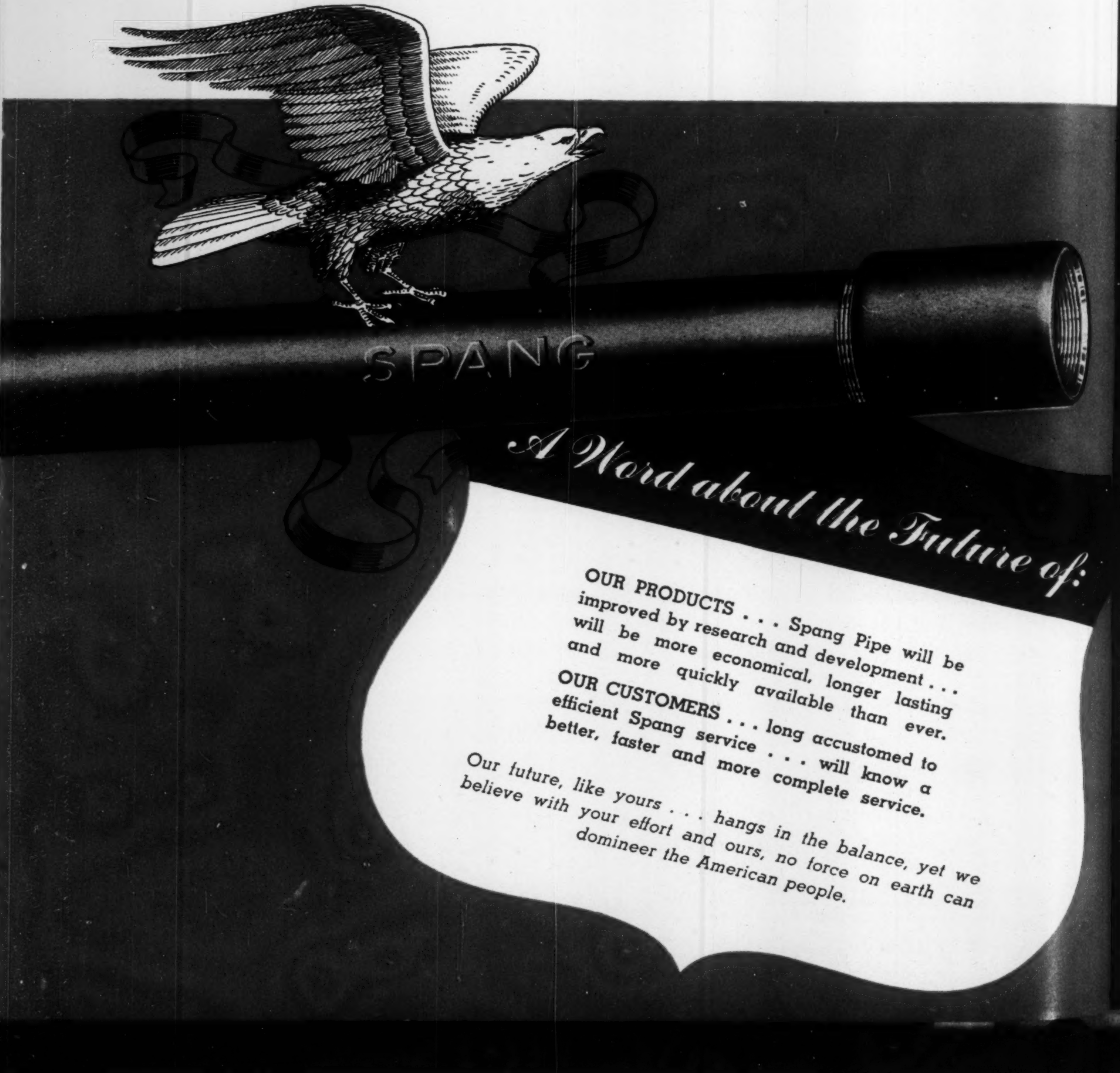

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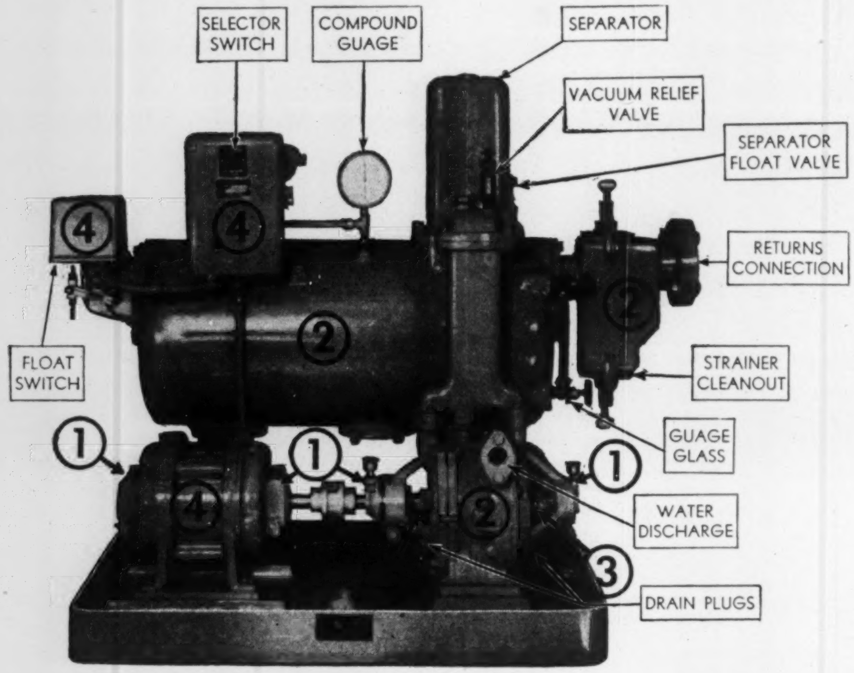
How to keep your Jennings Heating Pumps in First Class operating condition for Duration...



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are an important part of plant or building equipment. They cannot be easily replaced and must last for a long time. Jennings Pumps are designed for long life, and have extra capacity which will count when it may be difficult to maintain lines and equipment free from leaks, but to give full service they must have proper care.

Now that these pumps have been in service all winter on the heating system, they should have a thorough inspection. Special attention should be given to the bearings, as no wearing contact occurs in a Jennings Pump as long as the ball bearings are properly maintained.

In most cases Jennings Pumps will require only the check-up we have recommended. If additional adjustments or repairs are found to be necessary, they should be undertaken at once. Do not wait until both time and materials are running short. Nash Service Branches are established throughout the United States. If you do not know the address of the one nearest to you, write directly to this office. Nash Service is yours to command, NOW.

SERVICE POINTERS

❶ LUBRICATE PUMP AND MOTOR BEARINGS . . . Check bearings for end play, or radial play, and be sure they contain no grit, or foreign matter. If found to be clean and in good condition, fill with proper non-acid grease lubricant. If motor has oil lubricated bearings, flush out with fresh oil, and refill.

❷ CLEAN STRAINER AND STRAINER SCREEN . . . Clean strainer well and strainer screen. If screen is found to be worn, and has passed foreign matter, thoroughly flush both receiving tank and pump casing. Drain plugs are conveniently located . . . Damaged screen should be repaired or replaced at once.

❸ RENEW STUFFING BOX PACKING . . . Remove old packing and clean stuffing box. Fill with new loose-woven graphite-impregnated stuffing box packing, in individual rings. Be sure joints are staggered, and that ends butt properly without over-lapping. Do not finally tighten gland until pump is again in operation.

❹ HAVE MOTOR AND CONTROL INSPECTED . . . Have an experienced electrical service man go carefully over motor and control. Follow his recommendations and if repairs or replacements are indicated, have the work done immediately.

❺ CHECK OPERATOR'S INSTRUCTION BOOK . . . If operating instructions originally furnished with the pumps have been lost, write to us at once for a new copy for each pump. BE SURE TO FURNISH US THE TEST NUMBER OF EACH PUMP, as stamped on the pump name plate.

❻ IF PUMPS REQUIRE REPAIRS . . . and you do not know the address of our nearest Service Branch, notify us at once, and we will have our Service Engineer get in touch with you without delay . . . There are well equipped Nash Service Branches in most principal cities, ready to give you every possible help.

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THE PLAN to remove refrigeration equipment now installed for certain comfort air conditioning installations to make it available for war production has many interesting angles. Following are three articles giving practical information on the various points to be considered

---

Removal of Refrigeration Equipment from Existing Comfort Conditioning Systems for Use in War Production

SUMMARY—The successful removal of refrigeration equipment from existing comfort air conditioning systems in stores, office buildings, theaters, hotels, etc., for use in war production plants, will depend upon accurate application engineering and the care used in shipping and moving machinery. Present owners are offering their equipment on a voluntary and patriotic basis to help speed the war effort; wherever possible, the representatives of the war plant should make the removal operation as painless as possible, consistent with the all important time element, and thus permit a minimum of interference with the present owner's business. . . . Mr. Vroome gives here a most practical discussion of the many points to be considered, including suggested clauses for the various necessary contracts and data on how to maintain and operate the air handling equipment which is not removed in order to protect as much as possible the owner's investment. . . . Mr. Vroome is engineer in the mechanical department, design and appraisal division, Ebasco Services Inc.

A. E. Vroome Discusses Disconnecting, Removing, and Re-Installing Refrigeration Equipment, and How to Maintain and Operate the Equipment Not Removed

ANNOUNCEMENT made late in September by the War Production Board indicates that government agencies have requested owners of existing refrigeration machines voluntarily to sell their equipment to war industries. Executives of some department stores, office buildings, theaters, and hotels have already offered to sell the refrigeration plant, forming part of an air conditioning system, to the United States government or a war plant.

This memorandum gives information, from several viewpoints, relating to the disconnecting, removing, and re-installation of refrigerating equipment. Data are also presented on ways and means for maintaining the remainder of the air conditioning plant to protect the owner's investment and to provide partial summer service through adequate ventilation.

Interested Parties

The sale of equipment from an existing installation for application in a new location usually involves five parties; these include (1) purchaser, (2) seller, (3) supervising engineer, (4) dismantling contractor, and (5) erection contractor. Each of the foregoing has a special interest and responsibility which is discussed in the following paragraphs.

Purchaser—The purchaser may find it necessary to make three contracts: (1) with the seller for the sale of the actual equipment, (2) with the dismantling contractor for disconnecting and removal and loading for shipment, and (3) with the erection contractor. Anyone doing business in the construction field will recognize that there is considerable engineering detail involved in preparation of contract agreements, even when all erection and dismantling labor is done on a "cost plus" basis. This is the point where the supervising engineer enters the picture.

Supervising Engineer—The duties of the supervising engineer include survey of the existing plant and preparing specifications for equipment items to be purchased and for related contracts, involving mainly labor. The engineer should represent the war plant organization and be responsible for obtaining refrigeration machinery of suitable type, capacity, temperature range, and at a time to suit the construction schedule of the new plant. It is advisable that he should be the coordinator for all interested parties, and that he be given sufficient authority to make all necessary decisions to insure proper speed in the execution of the work. A competent engineer, preferably of the executive type, could handle a number of refrigerating plants in addition to taking care of other duties.

Contracts and Agreements

In preparing the contract between purchaser and seller, the actual equipment involved should be clearly stated. For example, a clause might read as follows:

> "The refrigerating machinery included in this agreement is to be 'as is' and 'where is' and shall include the centrifugal compressor, gear box, electric driving motor, flexible couplings, speed controller and motor starter, water cooler, refrigerant pump, purge device, control panel and gages, together with all safety devices such as low temperature cutout, high pressure cutout, overspeed trip, fusible plug, and all relief valves. The refrigerant condenser together with its accessories shall be included. All bedplates, cradles, and structural supports of steel or cast iron shall be included."

In some instances the electric current characteristics of the existing motor may not be suitable, such as a d-c motor in present plant whereas the new plant is a-c. In this case the motor and control would not be purchased.

Another item to be included in the requisition for each machine is a statement of capacity, substantially as follows:

> "It is agreed that the manufacturer's rating of the refrigerating plant as installed in John Doe's department store is ——— tons of refrigeration when cooling ——— gallons of water per minute from an entering temperature of ——— F to a leaving temperature of ——— F when the condenser is supplied with ——— gpm of cooling water at ——— F entering and ——— F leaving. Pressure drops through water cooler and condenser, with clean tubes, are ——— psi and ——— psi respectively, when handling the stated amounts of water."

Other items which the supervising engineer must investigate are foundation drawings, spare parts on hand, erection and operating instructions, operating log sheets, maintenance work done in the past and expected in the future, condition of the tubes in water cooler and condenser, and specifications and quantity of lubricating oil and of refrigerant required for normal charge. In some instances, it may be desirable to include the refrigerant; in this event the necessary containers must be secured for shipment. Data on normal refrigerant loss should be obtained and if it appears excessive, precautions taken to renew seal, packing, etc., when the machine is erected in the new location.

Dismantling Equipment

Another duty of the supervising engineer is preparation of the requisition for dismantling equipment and moving it to the railroad siding, this work to be done by a local contractor satisfactory to the original owner of the refrigerating plant. The order should specify clearly the scope of work called for, the time, both as to dates and time of day (during normal business hours or after), insurance or responsibility for any damage to refrigeration machine and other building equipment. The following clause might be used in the dismantling contract:

> "The contractor shall disconnect, remove from the building and load on railroad cars, the following refrigerating equipment, compressor, water cooler, refrigerant condenser, etc. (list all items). A manufacturer's representative or erection engineer shall be present and direct the work. Standard rate for this service is ——— per day plus living and traveling expense. Cost of said services shall be paid for by the contractor.
>
> "The contractor shall prepare all parts for shipment, providing heavy timber skids and cradles for the large pieces of equipment and crates with waterproof paper over apparatus for accessories. All small parts shall be boxed, tagged, and numbered. Machined surfaces shall be covered with heavy grease or other suitable rustproof coating and all flanges and pipe connections protected with heavy wooden covers or plugs.
>
> "All pipes shall be suitably closed and all foundation bolts left in place and protected against damage. Should it be necessary to move other equipment to permit passage of refrigerating machine or to cut masonry construction, same shall be done by the contractor at a time satisfactory to owners, and restored to original condition at the completion of the work."

Partial Operation of Air System

Thorough investigation should be made of various methods for using the air handling portion of the air conditioning system after all or part of the refrigerating plant is removed, since it will usually be possible to use the air circulating fans after the refrigerating plant has been dismantled. Usual practice in proportioning the amount of outdoor air and recirculated air is to provide from 10 to 25 per cent outdoor air and to recirculate the remainder. This procedure should, if feasible, be changed to permit supplying up to 100 per cent outdoor air. In some instances, this may cause complication in the size of outdoor air ducts and the general arrangement of air filtering equipment. It will also place a greater load on the air filters, requiring them to be cleaned or serviced more frequently.

The success of utilizing the air handling equipment without refrigerating machinery will depend largely upon the geographical location of the plant, internal heat load, and occupancy. Obviously, an air conditioned store in Boston will have lower outdoor air temperatures than a similar store in New Orleans. Therefore, greater success on straight ventilation may be expected from stores and office buildings in the New England states offering their refrigerating equipment to war plants than can be expected under similar conditions in areas farther south.

Protection of Equipment if Shut Down

In the event that a building owner does not elect to operate his fan equipment, certain precautions should be taken to protect the fans, motors, coils, and similar items during the shutdown period for the duration. It is suggested that all motor equipment be cleaned and then covered with cardboard cartons or wrapped in heavy paper to prevent dust, dirt, and moisture entering the windings. Fan wheels and casings should be inspected and painted, after removing all corrosion and dirt. Air filters should be thoroughly cleaned, removed, and stored in a dry place. Air cooling coils should have the outside surfaces cleaned, and if of the water type, should be drained and blown out with compressed air, then protected by suitable covering.

Partial Air Conditioning

There will be cases where part of a refrigerating plant may be removed for governmental use and the remainder be available for partial cooling service. In many buildings there are spaces which definitely require air conditioning to provide livable conditions for occupants. An illustration is a bank vault below the grade level, which is entirely dependent upon mechanical ventilation and air conditioning. In spaces on upper floors it is usually possible to open windows and provide reasonable air circulation either from outdoor air movement or by circulating fans inside the building. Each building owner should determine which of the several areas in his building has the greatest need for air conditioning service and arrange his equipment and piping to meet this requirement.

Re-installation for War Industry

The war plant in which the refrigerating machine is to be placed should fix the time of delivery for the machinery. Proper foundations should be built and ready to receive the refrigerating machine when it is moved from railroad car to the plant. Either original foundation drawings or accurate field measurements must be secured before foundations can be built. The engineer should determine at the start of the project, all clearance requirements for removal of tubes in water cooler and condenser, headroom to lift the heads on compressor, weight of entire unit, and position of new building column foundations with relation to that required by the refrigerating machine. If possible, the same manufacturer's erection engineer who supervised the dismantling of the equipment should be employed to take charge of the erection in the war plant.

The action of the War Production Board in asking department stores, theaters, hotels, and office buildings with air conditioning equipment above 100 hp horsepower to "volunteer" such equipment for "active duty" in war production factories should serve to eliminate from the public mind the misconception of air conditioning merely as a "comfort" industry, says Dr. Willis H. Carrier, chairman of the board of Carrier Corp.

"It is seldom recognized that air conditioning had its inception in industry as a production tool," Dr. Carrier said. "Not only is low temperature refrigeration equipment essential to the production of certain types of synthetic rubber, as pointed out by Donald Nelson in his request that stores and others 'volunteer' their equipment, but air conditioning equipment serves a myriad of other war production requirements."

RELEASING REFRIGERATION FOR WAR PRODUCTION

The movement to release refrigeration equipment now used in comfort air conditioning systems for war production is very interesting.

There are a large number of comfort air conditioning systems throughout the country that have capacity in excess of that required to meet design conditions. There are also a large number of systems that have been over-designed and have a capacity greater than required to maintain a minimum degree of comfort. On these installations, especially where there are two or more compressors, the loss of one-third to one-half the total capacity should not seriously affect the usefulness of the system.

The elimination of all refrigeration from a well designed air conditioning system having the capacity to handle 100 per cent outside air still leaves a good ventilating system capable of producing considerable relief to the conditioned area. Those systems designed with an air washer type exchanger, particularly in localities having low outdoor wet bulbs, will have additional effectiveness because of the possibility of employing evaporative cooling on some occasions.

I do not believe that the movement can be promoted successfully on a national basis; it should be handled locally. A department store owner in Pittsburgh would not be interested if an owner in Cleveland agreed to give up all or part of his refrigeration, but he would be interested if all the department stores in Pittsburgh agreed to give up a certain percentage of theirs.

There is no particular problem involved in the removal of the refrigeration portion of an air conditioning system, where the refrigeration is used only for summer operation. The air and refrigerant circuits are self contained and separate and the entire refrigeration system should be removable without affecting the air circuit.—A. J. Lawless, consulting engineer.

Refrigeration for Comfort Conditioning To Be Drafted for Essential Production

E. P. Heckel Tells Why Existing Machines Are Needed for Synthetic Rubber Plants, Explains Effect on Present Installations

SUMMARY—Government speakers announced recently that refrigeration machines, condensers, and prime movers now installed for air conditioning, stores, theaters, hotels, office buildings, etc., are to be taken for use in synthetic rubber and other essential war production (see p. 606, October, 1942, HPAC). Machines of 100 hp capacity and larger, not over five years old, are wanted. . . . Mr. Heckel, E. P. Heckel & Associates, a member of HPAC's board of consulting & contributing editors, explains here the need for "drafting" such equipment for war work and its effect on building owners. He suggests the need for expert assistance in disassembly and reassembly, and the importance of getting the original drawings. . . . He concludes with suggestions on operation of the remaining equipment, and presents an estimate of the refrigeration tonnage required for synthetic rubber production

DEPARTMENT STORES, chain stores, hotels, theaters, office buildings and many other users of air conditioning and refrigeration can contribute to our war effort by voluntarily offering for use by our government their existing refrigeration machines, which are needed in the production of synthetic rubber and other essential civilian and war materials.

Time is short, and the labor manhours to produce all of the required refrigeration capacity needed in the form of new equipment necessary to do this job are limited.

Obviously, those who are more familiar with the situation that confronts us believe most sincerely that the contribution by owners of existing refrigeration machines of 100 hp capacity and over to do this job is not only patriotic but will materially aid in the preservation of our American way of life and freedom. The acquisition by the government of existing refrigeration machines will speed up tremendously the processing of production capacities for war essentials.

Steel and iron and many other critical metals are needed in the production and fabrication of parts for refrigeration machines, many of which are now unobtainable or extremely difficult to procure. It can be conservatively estimated that the weight of a large refrigeration machine is some 500 lb per ton of capacity. Steel tubes are now used in place of the copper or brass with which most existing refrigeration machines are equipped and which have better heat transfer characteristics. After the satisfactory conclusion of the war, when we again can expect to return to business as usual and normal life, it seems reasonably certain that the store, hotel, theater, and office building owners who have contributed their machines for the present emergency needs will be able to have their old machines replaced by new ones with all of the added refinement of parts, etc., developed since their original machines were installed. This will have the advantage of creating a "leveling out" process to keep the manufacture of refrigeration machines up to capacity after the war.

Right now, however, with the extremely huge refrigeration capacity that is required and which is being contributed in the form of existing machines, thus saving metals and time, we still have the important job of moving the disassembled machine from the old foundations upon which it has rested to the new location for reassembling and for operation. This must be done with skill and precision, by mechanics familiar with such procedure. Here is where the aid of the original manufacturer of the machine is highly important. He has and should provide not only copies of the original blueprints of the machine, but also of the foundation layouts, and all layouts of auxiliary service connections such as steam, electric starters, condensing water connections, sewer and drain lines—in fact copies of all drawings pertaining to the installation and operation of the machine. He should be called upon to furnish a superintendent thoroughly familiar with the peculiar needs of his special equipment, to supervise such work. Since it may be expected that many different types and makes of machines will be involved, it seems reasonable that all manufacturers of refrigeration machines will need to render this important service.

What about the owner who has contributed his refrigeration machine to serve in this emergency? How will he be affected?

He—more than anyone else—is fully aware and is more accurately

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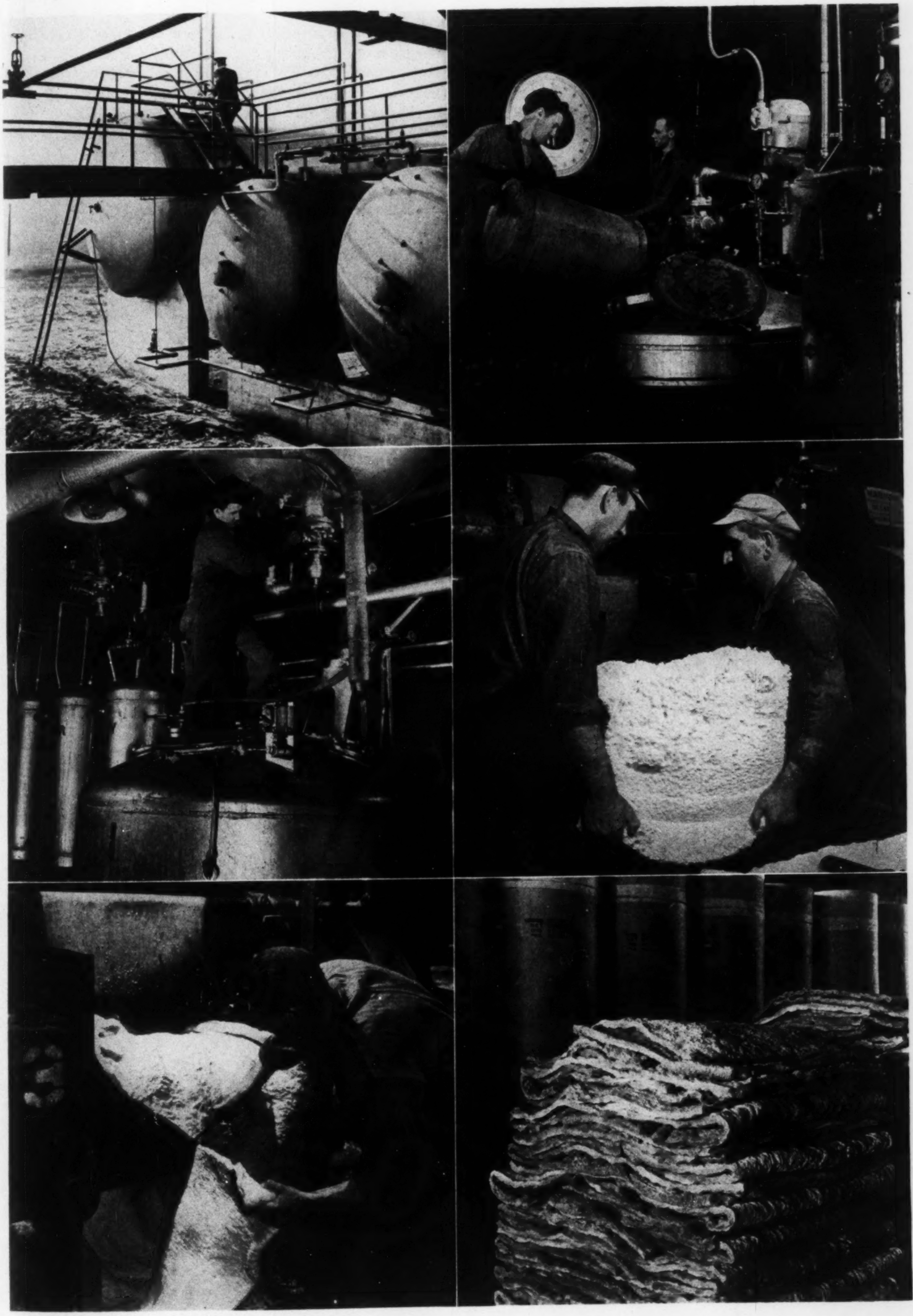
SYNTHETIC RUBBER MANUFACTURING

(See opposite page)

Top left—**These tanks contain raw materials for synthetic rubber: vegetable fats and butadiene, a highly volatile gas cracked from crude petroleum. . . .** ***Top right***—**Carefully selected soap is used in the making of solutions used during the manufacture of synthetic rubber. This worker is mixing a solution in a nickel lined tank. . . .** ***Middle left***—**This worker is standing on a blowdown tank, about ready to tap a polymerizer tank. The latex made in the latter tank is stabilized in the blowdowns. A large blending tank increases the uniformity of the synthetic solutions by mixing the contents of seven or eight of these tanks. . . .** ***Middle right***—**Removing rubber crumbs from the perforated boxes below the coagulating and extractor tanks. The rubber crumb slurry flows into the boxes from the tanks, and the water previously added to dissolve the soap in the solution runs off through the perforations in the box. The rubber will now be prepared for the wash mill. . . .** ***Bottom left***—**A sheet of synthetic rubber coming off a rolling mill. . . .** ***Bottom right***—**Finished synthetic rubber ready for shipment in large paper drums holding 200 lb each. . . . (Office of War Information photos by Palmer taken at a B. F. Goodrich Co. plant)**



informed of what such service disruption means to him. The inability to maintain comfort conditions in his public spaces, sales areas, hotel rooms, offices or theaters during the summer months is a real disadvantage. He is willing to accept such disruption for a short time or a few years in preference to such acceptance for all time, should we lose the war. He will still have his air handling system intact with which to provide comfort conditions during the winter months and throughout the non-cooling demand periods of the year. Unfortunately, many such fan system capacities have been basically designed to function best with high efficiency when operated in conjunction with the refrigeration machines during the warm summer months. With the refrigeration machine capacity unavailable for cooling air, owners may find it desirable to provide more outside air and more exhaust to aid and better the prevailing conditions.

A large percentage of the owner's original investment for air conditioning equipment will remain in service even after the now existing refrigeration machine has been removed. For this he is presumably receiving a fair compensation based on the original cost, less a reasonable deduction covering charged off amortization.

It is impossible to estimate with any great degree of accuracy the amount of refrigeration which will be required for synthetic rubber production. Much of the data aren't available—at least to the public—and numerous assumptions must be made.

However, let us suppose that a synthetic rubber production of 650,000 tons annually is desired; dividing by 365, this is, say, 1800 tons of synthetic rubber per day or 1.25 tons per minute (assuming a 24 hr day). This is equal to 2500 lb per min of finished raw product.

Assume further that the finished product is equal to 10 per cent of the original mass, and that cooling of the original mass to minus 100 F is necessary in the manufacturing process. Thus, we have 25,000 lb per min to be cooled to minus 100 F (2500 lb of finished product and 22,500 lb of unusable byproduct).

Considering the specific heat to be one, and cooling through a range from plus 80 F to minus 100 F, requires a cooling capacity of 180 × 1 × 25,000 or 4,500,000 Btu per min. As one ton of refrigeration is defined as 200 Btu per min, this is equivalent to 22,500 tons of refrigeration, based on low temperature operation and without any allowance for standby capacities or cooling losses. Since these low temperatures cannot be attained with single stage refrigeration and available condensing water at ordinary temperatures, dual stage refrigeration and cascading must be employed, plus refrigeration to accomplish the condensing for the machines operating to produce the base low temperature. Thus, the figure of 22,500 tons of refrigeration capacity would have to be materially increased in order to compare it with the tonnage capacities of refrigeration machines operating in the relatively high temperature range for air conditioning work.

FAN LAWS — ARRANGED FOR READY REFERENCE

(From the articles by William Goodman in Heating, Piping & Air Conditioning, August, September, and October, 1942)

	VARIATION IN:			
	VOLUME	PRESSURE (Static or Total)	HORSEPOWER	SPEED
SPEED CHANGED	$\frac{V_2}{V_1}=\frac{N_2}{N_1}$ [Eq. 3, p. 489, Aug.]	$\frac{p_2}{p_1}=\left(\frac{N_2}{N_1}\right)^2$ [Eq. 16, p. 489, Aug.]	$\frac{(hp)_2}{(hp)_1}=\left(\frac{N_2}{N_1}\right)^3$ [Eq. 22, p. 489, Aug.]	
DIAMETER CHANGED	$\frac{V_2}{V_1}=\left(\frac{D_2}{D_1}\right)^3$ [Eq. 4, p. 489, Aug.]	$\frac{p_2}{p_1}=\left(\frac{D_2}{D_1}\right)^2$ [Eq. 17, p. 489, Aug.]	$\frac{(hp)_2}{(hp)_1}=\left(\frac{D_2}{D_1}\right)^5$ [Eq. 23, p. 490, Aug.]	
SPEED AND DIAMETER CHANGED	$\frac{V_2}{V_1}=\left(\frac{N_2}{N_1}\right)\left(\frac{D_2}{D_1}\right)^3$ [Eq. 9, p. 489, Aug.]	$\frac{p_2}{p_1}=\left(\frac{N_2 D_2}{N_1 D_1}\right)^2$ [Eq. 15, p. 489, Aug.]	$\frac{(hp)_2}{(hp)_1}=\left(\frac{N_2}{N_1}\right)^3\left(\frac{D_2}{D_1}\right)^5$ [Eq. 21, p. 489, Aug.]	
STATIC PRESSURE AND OUTLET VELOCITY CONSTANT	$\frac{V_2}{V_1}=\left(\frac{D_2}{D_1}\right)^2$ [Eq. 29, p. 535, Sept.]		$\frac{(hp)_2}{(hp)_1}=\left(\frac{D_2}{D_1}\right)^2$ [Eq. 25, p. 535, Sept.]	$\frac{N_2}{N_1}=\frac{D_1}{D_2}$ [Eq. 24, p. 535, Sept.]
DENSITY CHANGED; SPEED CONSTANT		$\frac{p_2}{p_1}=\frac{d_2}{d_1}$ [Eq. 43, p. 617, Oct.]	$\frac{(hp)_2}{(hp)_1}=\frac{d_2}{d_1}$ [Eq. 44, p. 618, Oct.]	
DENSITY CHANGED; SPEED CHANGED	$\frac{V_2}{V_1}=\frac{d_1}{d_2}$ [Eq. 35, p. 536, Sept.]	$\frac{p_2}{p_1}=\frac{d_1}{d_2}$ [Eq. 36, p. 537, Sept.]	$\frac{(hp)_2}{(hp)_1}=\left(\frac{d_1}{d_2}\right)^2$ [Eq. 40, p. 617, Oct.]	$\frac{N_2}{N_1}=\frac{d_1}{d_2}$ [Eq. 32, p. 536, Sept.]

SYMBOLS

D = diameter of fan wheel, inches
d = density of air, pounds per cubic foot
(hp) = horsepower
N = fan speed, revolutions per minute
p = pressure, inches of water
V = volume of air, cubic feet per minute

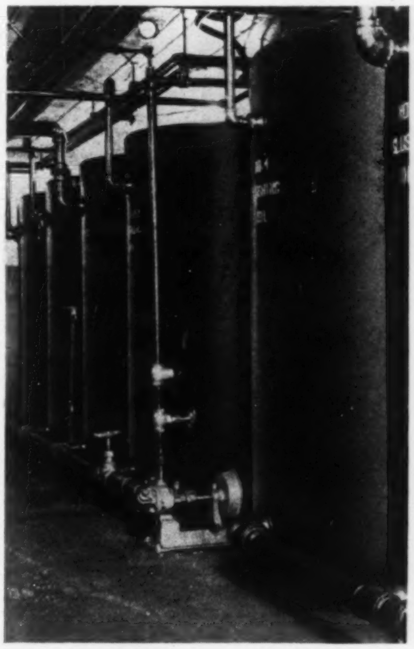
New Oil Distribution System Cuts Consumption, Helps Maintenance

Centralized Storage and Pipe Line Distribution of Oils and Coolants at Drawn Steel Plant Has Number of Advantages

Summary — **Eight different petroleum products are pumped in separate piping systems to individual machines and points of use in a plant producing bars, wire, precision shafting, and special shapes of cold finished steel. The piping, pumping, settling, filtering, and reservoir equipment are described here in detail. . . . Among the advantages of the centralized system over the former method are reduced consumption, improved machine maintenance, greater uniformity of the oils, lower temperature of returning coolants, and cleanliness**

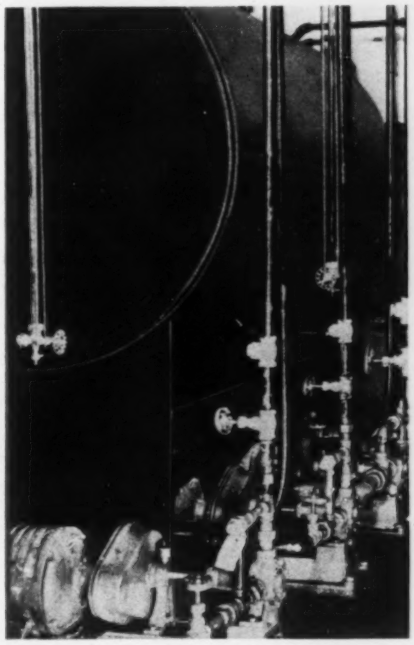
A NEW oil distribution system recently completed at a plant of a division of Republic Steel Corp. has improved machine maintenance and cut oil consumption and container handling cost to such an extent that plans are already under way for similar installations at four other plants of the company. Altogether, eight different petroleum products are pumped in separate pipe lines to individual machines and points of use throughout the plant, which produces bars, wire, precision shafting, and special shapes of cold finished steel. Turbine oil, kerosene, cylinder oil, "Polarine" oil, soluble oil, drawing oil, and two grades of slushing oil (light and heavy) are handled in this fashion. The soluble oil is used as a coolant for metal saws, lathes, and grinders. The drawing oil is required for lubricating the dies used in cold drawing bars. The two grades of slushing oil are used for spraying a protective film on finished stock as it is loaded into railroad cars on the depressed siding which runs into the plant shipping department.

The eight oil receiving lines are enclosed in this steel and concrete compartment for protection against fire, theft, and other hazards



A group of 280 gal tanks in an underground vault is used for the storage of eight different kinds of oil required in plant processes

Delivered to the plant by truck, each grade of oil is allowed to flow by gravity into its individual storage tank in a ventilated underground vault through 2 in. filling pipes brought up inside an accessible concrete enclosure beside the plant building. Weatherproof hinged covers on each oil line and locks on the sheet steel cover of the enclosure protect against fire, theft, or other hazards.



One of the storage tanks is divided into three sections and is mounted horizontally, on a tank saddle. Pumps are driven by explosionproof motors

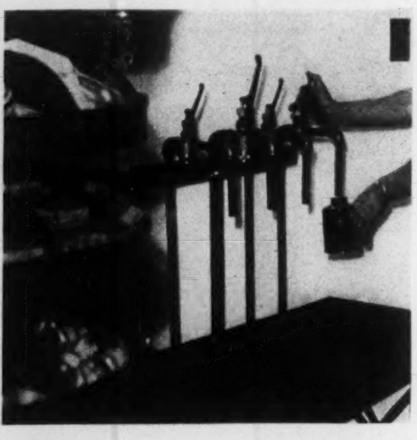
The storage vault houses eleven 280 gal tanks, installed vertically side by side, and a twelfth, also of 280 gal total capacity, which has been divided into three sections of equal size and mounted horizontally on a tank saddle of all-welded steel construction.

Such large quantities of slushing oil are consumed that eight of the vertical tanks, connected together in batteries of four tanks each, have been provided to handle the hundreds of gallons of the two grades used per month.

Turbine oil, drawing oil, and soluble oil are each provided with separate storage tanks. Consumption of "Polarine," kerosene, and cylinder oil is in each case enough smaller that adequate reserves can be handled by allocating to each a section of the three section tank.

Gear Pumps

Gear pumps, equipped with relief valves on the pump heads, are used on all eight oil lines fanning out from the central storage vault. They are driven by explosionproof, 3 phase, 220 volt, 60 cycle motors. On the soluble oil installation and on the drawing oil installation, where an interruption to supply would seriously inconvenience plant operations, two such pumps, one a standby, have been installed on each set of storage tanks.



Turbine oil, kerosene, cylinder oil, and "Polarine" are dispensed in small quantities from these lines brought to the stockroom from the underground storage vault

In addition to the ordinary control switches on the motor circuits, mercury switches were installed on the slushing oil pump handling the light and on the main and standby pumps for the soluble oil to take care of the situation created by a tendency of these oils to leak past the diaphragms of ordinary pressure switches and render them inoperative. Motor starter switches and pressure gages were grouped on the wall immediately outside the storage vault.

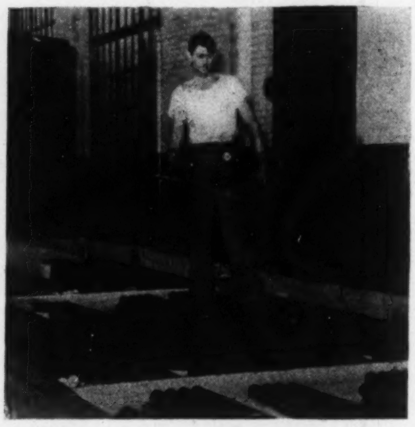
Ventilation of the storage vault is provided by 4 in. pipe sections at both ends of the room which exhaust to the outside atmosphere. One vent line extends down to within 4 in. of the floor; the other opens directly above the tanks at the ceiling level. Between the two it is possible to avoid a dangerous accumulation of fumes at any level.

From the centralized storage room, iron pipe lines, carried close to the basement ceiling, and of varying sizes depending upon the viscosity of the oil handled and the length of travel involved, carry the different oils to four principal points in the plant: The store room, the shipping department, and to settling tanks for separate drawing oil and soluble oil distribution systems.

All oil lines pass from the underground storage vault through a short insulated tunnel provided with steam heating coils to insure the maintenance of at least 70 F temperature, even when outside temperatures fall as low as zero. This tunnel carries the pipe lines into the main section of the plant basement.

Identification

Four lines carry turbine oil, kerosene, cylinder oil, and "Polarine" oil to the store room where they are dispensed in small quantities. Kerosene and "Polarine" are pumped under 20 psi pressure. This is increased to 40 psi for the turbine oil and to 60 psi for the cylinder oil. For identification these pipes are painted as follows: Turbine oil — yellow; kerosene — red; cylinder oil—blue; and "Polarine" oil—aluminum.



Regulation spray gun equipment on 25 ft lengths of hose is used for spraying finished steel with slushing oil

The slushing oils are delivered to the shipping department through 400 ft supply lines from the storage room and carried along the depressed railroad loading bay in two separate lines. One inch pipe, painted brown, is used to deliver the heavy slushing oil under 80 psi pressure; ¾ in. pipe, painted black, handles the light slushing oil at from 20 to 40 psi pressure. A 1¼ in. line (gray) carries air under 20 psi pressure from air compressors situated at the other end of the shipping department and parallels the two oil lines for over 200 ft along the track.

Eleven stations are provided along the track loading zone where spray gun equipment on 25 ft lengths of hose can be attached to either oil line and the finished steel sprayed as it is loaded into the railroad cars. Oil and air outlets are equipped with fast coupling connectors, consisting of a socket and plug, to speed attaching and removing the spray units.

Inasmuch as some finished steel is shipped by truck from the plant, a portable slushing outfit was also provided. It consists of a small, mobile unit on which is mounted small, self-contained pumping and air compressor equipment. Separate 54 gal drums are provided to carry either light or heavy slushing oil which is sprayed from this portable equipment under 32 psi air pressure.

Soluble Oil

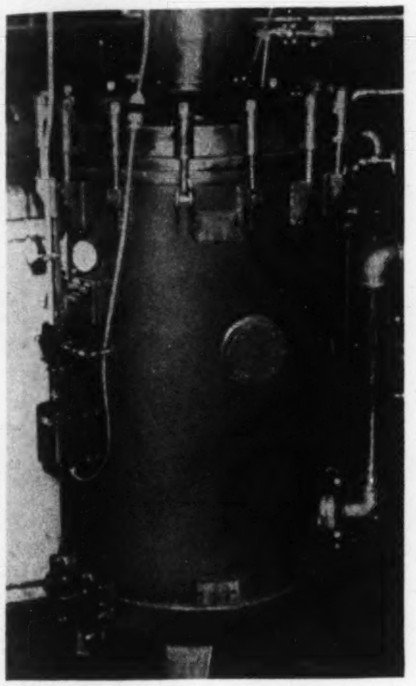
Lathes, grinders, cutoffs, and saws require a quantity of soluble oil in their operation. To provide it, a self-contained and self-cleaning distribution system has been set up with supply and return lines to handle the soluble oil to and from all equipment of this nature in the plant.

Heart of the system is a 12 ft diameter, open settling tank in the basement of the plant. The soluble oil is pumped from storage through 178 ft of 1¼ in. pipe to a small measuring tank immediately adjacent to and elevated above the settling tank. It is mixed for use as a coolant in proportions of 10 to

Oil and air outlets are equipped with fast coupling connectors, consisting of a socket and plug, to speed attaching and removing spray units

one with water which is introduced into the settling tank directly from a plant water main. From the settling tank the coolant is pumped under 40 psi pressure through a self-cleaning coolant filter to the machines on the floor above. It returns by gravity to the settling tank.

The abrasive particles which are picked up in the coolant oil would produce excessive wear on a gear pump, and consequently a centrifugal pump is used for this assignment. The pump has a capacity of 225 gpm and is driven by a 10 hp, 220 volt, 60 cycle, 3 phase, explosionproof motor.



This tank in the plant basement is a part of the soluble oil distribution system

The two cylindrical filtering units contained in the coolant filter are connected through gearing to a motor driven speed reduction unit. They turn only when cleaning is necessary and then at 6 rpm, thereby minimizing power consumption and wear and tear on equipment. The driving motor is controlled by a differential pressure switch which operates only when the differential pressure increases to a predetermined point. When the filtering units are cleaned, which usually requires only a fraction of a turn, the differential pressure drops and the motor is stopped. A manually operated switch is also provided to permit the motor to be started and stopped at will.

An exceptionally large sump is provided below the filtering units to permit lower filtration cycles without necessity of shutdowns for cleaning. The discharge line from the filter has a 3 in. relief valve, 10 to 300 psi pressure range. This is set at 45 psi pressure and is connected so that it can bypass oil into the return line to the settling tank. Maximum capacity of the filter is 400 gpm.

Settling Tanks

At each machine, whether grinder or lathe, there is a primary settling basin to remove as many as possible of the metal particles, chips, etc, which would otherwise be returned through the piping to the main settling tank. The carborundum bonding cement or sizing used in the composition of grinder wheels, for example, would quickly stop up even a large size pipe if permitted to pass through in any quantity. The individual settling basins at each machine, therefore, consist of a wire basket suspended in a sheet metal basin. The coolant is discharged from the machine into the metal basket which holds back most, if not all, of the objectionable foreign matter from entering the return line to the settling tank.

At all junction points in the return lines from these basins, 45 deg Y bends in the direction of flow are used to minimize further any possible clogging of the lines. Likewise, all return lines have a minimum number of bends and fittings and were kept as nearly as possible in a straight line.

At the settling tank, the return line empties into a screen basket of 100 x 100 mesh welded to and reinforced on the outside by a perforated shell having 1 in. holes on $\frac{11}{16}$ in. centers or the equivalent of 48 per cent open area. Sixteen inches in diameter and 3 ft in overall depth, this basket arrangement permits the oil to dissipate its initial turbulence before overflowing into the settling tank proper.

So that the settling tank can be cleaned at frequent intervals, yet without interrupting operations, the bottom was divided into three sections by 3 ft high baffles. Since the depth of the coolant in the tank is generally carried at about 30 in., this permits discharging the returning coolant into one section of the tank while cleaning another.

The longest and most extensive of the oil circulating arrangements in the plant, the delivery side for the soluble oil, involves 680 ft of pipe, graduated in size from 3 to ¾ in. On the return side, 630 ft of pipe from 4 to 1 in. in size was used. For identification all soluble oil lines are painted white.

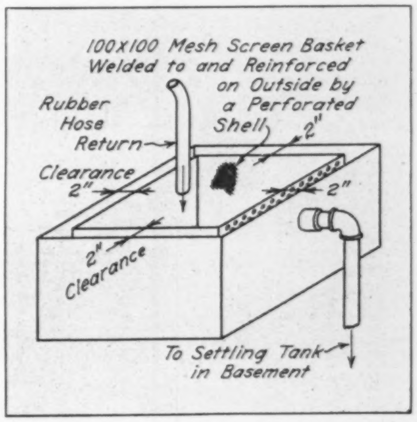
Drawing Oil

Basically the same elements are required in the distribution system handling the drawing oil as are needed for the distribution of the soluble oil. The character of foreign matter picked up in the oil, however, made some innovations necessary to insure the proper functioning of the system, which consists of three sections:

260 ft of 1 in. make up line from oil storage vaults to two settling tanks.

130 ft of supply line, divided almost equally between 1 in. and ¾ in. pipe, which delivers the oil to the drawbenches.

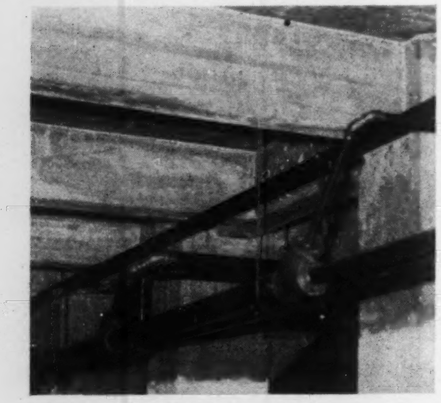
A 125 ft return line, much of it 1½ in. pipe, the balance 1 in., which deposits the drawing oil in either one of two settling tanks in the plant basement.


100x100 Mesh Screen Basket Welded to and Reinforced on Outside by a Perforated Shell
Rubber Hose Return
2"
Clearance 2"
2"
2" Clearance
To Settling Tank in Basement

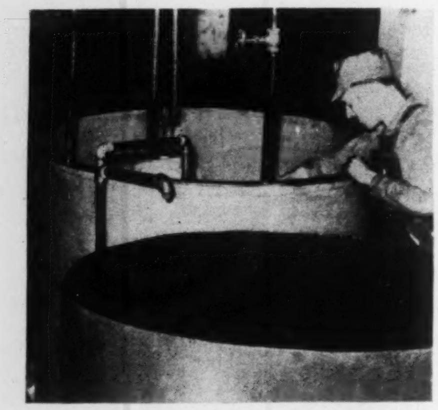
Settling basin for grinders and turning machines

All this piping is painted green for identification.

The number of machines to be served and the length of travel of the drawing oil are much less than in the soluble oil set up. At the same time the quantity of oil to be kept in circulation is sizeable and it was found convenient to use two settling tanks, rather than one, to facilitate cleaning operations; while one tank is being used the other can be cleaned. A gear pump of 10 gpm capacity is equipped with a relief valve on the head and driven by a 1 hp, 220 volt, 3 phase, 60 cycle, explosionproof motor. It operates continuously to maintain the oil pressure at approximately 40 psi.



Forty-five degree Y bends in the direction of flow are used to minimize clogging of return pipe lines



Two settling tanks, rather than one, are used to facilitate cleaning of the drawing oil. Piping equipment shown here was installed in such fashion that it can easily be swung into position over the second tank while the first one is being cleaned

The relief valve on the head of the pump has been found sufficient to hold the oil pressure steady at this value.

The discharge line from the pump passes all the oil through a pressure liquid type filter using an absorbent filter cartridge. However, steel wool was substituted as the filtering medium in place of the usual cotton waste, wool, excelsior, or asbestos. Steel bars in the manufacturing process are pickled to remove scale, then dipped into wash water to remove acid. From the wash water they are put through a lime solution at 220 F. This leaves a fine coat of lime deposit on the bars which, picked up in the drawing oil, was found to fill up a cotton waste filter very quickly. No. 3 steel wool was substituted as a filtering medium and has been found to be more satisfactory. While the steel wool permits the lime particles to pass through, this is entirely satisfactory since lime is in itself a lubricant and need not be removed from the oil as it does not damage the drawing dies. As in the piping circuit for the soluble oil distribution, the discharge line from the drawing oil filter is equipped with a 1 in. relief valve, 10 to 300 psi pressure range, and bypasses into the return line to the two settling tanks.

Advantages of the System

All of the oil distribution circuits in the new system have been functioning satisfactorily from their inception a few months ago. The centralizing of all oil handling has introduced a number of advantages over and above the obvious ones. This is particularly true in connection with the distribution of soluble oil. Previously, soluble oil was distributed around the plant in individual barrels. Used as a coolant for the metal saws, lathes, and grinders, the oil was mixed with water as needed. Any one of several individuals handled these mixing operations with the result that the supply varied widely in consistency and was frequently not so satisfactorily proportioned as was wanted. Distributing the soluble oil from a centralized system, on the other hand, has made it possible to put the mixing into the hands of one man. This makes for absolute uniformity of results and for proper consistency at all times.

The establishment of a large central reservoir of soluble oil has also eliminated one of the problems always presented with small self-contained systems. The many times larger quantity of soluble oil in circulation in the new system gives the oil a chance to cool off before being returned to the machines. In the small self-contained systems of individual machines there is always a chance of the coolant getting too hot and permitting the tools, the machine, or the work itself to be damaged through the development of excessive temperatures.

Likewise, with centralized control of the supplies of coolant and drawing oils the pressure at which these oils are delivered to the points of use can be kept more uniform. Generally, too, the oils as they are returned to service are much cleaner than would be possible if they were handled in any other way.

FIND FUEL VARIATIONS AFFECT BOILER TESTS

Engineering representatives of members of the Institute of Boiler and Radiator Manufacturers recently met at Johnstown, Pa., to discuss certain problems in connection with the testing and rating of boilers of approximately 40 in. grate width. John P. Magos, chairman of the technical committee, presided.

In opening the meeting, he stated that the committee had deemed it advisable to have tests conducted in one laboratory on several boilers of approximately 40 in. grate width in order to determine whether the provisions of the IBR code should be revised in any way so as to obtain results on boilers over 30 in. grate width which are accurate beyond any question. Tests were conducted with two boilers during the course of the meeting. As a result, it was decided that the only modification which might improve the code is to tighten up on the analysis of fuel used in tests.

LIMITATION ORDER L-38 RECENTLY AMENDED

Production of additional types of air conditioning and commercial refrigeration equipment was prohibited by the War Production Board with the issuance of general limitation order L-38 as amended September 26. The amended order also allowed sales of specified types of equipment in the hands of dealers and manufacturers, in addition to equipment released for sale by the previously amended order.

Production of evaporative coolers with a capacity of 2000 cfm of air or less was cut off. Evaporative coolers rated over 2000 cfm can be manufactured only for preferred orders.

CHECK LIST FOR WARTIME HEATING ECONOMY

By Eph. A. Tyler, Rental Manager, First National Bank Building, St. Paul, and Member of HPAC's Board of Consulting & Contributing Editors

The principles of heating economy are simple: (1) Provide sufficient steam to offset outside temperatures only. (2) Stop all leaks of good heated air from the building. (3) Cut down inside building temperatures as low as possible when building is not occupied by the working force. . . . The following check list will help in finding and correcting wasteful operation:

DISTRIBUTION

1) Controls in perfect working order..........——
2) Leaks in mains, risers, and runs all stopped..——
3) Vacuum pumps working 100 per cent....——
4) Drip traps cleaned (monthly) (quarterly) (annually) as needed......................——
5) Radiator traps checked monthly...........——
6) Radiator traps all cleaned annually; worn out seats and thermostatic elements replaced....——
7) Reducing valves working properly..........——
8) Check insulation on all feed lines and risers, all covered completely ....——
9) Radiation all balanced...................——
10) Longer hour zones piped separately........——
11) Heat in condensate recovered.............——
12) Valves for night shutoff and controls for reducing valves convenient and accessible......——
13) All radiators checked to be sure of no obstruction to free circulation of air around them..——
14) Hot water tanks below 140 F..............——
15) Controls for heating outside water tanks set for no heat above 32 F and lowest possible temperature ....——
16) If piping system "sludge-dopey," to be cleaned at first opportunity......................——
17) Circulation free and rapid................——

BUILDING LEAKS

1) Entrance swing doors fit snugly............——
2) Rubbers and felts on revolving doors repaired and snugly fitted..........................——
3) Defective weatherstripping on windows repaired ....——
4) Defective window frame calking repaired...——
5) Windows not needed bricked up or blanked off with insulating material...............——
6) All penthouse doors and windows fit snugly. (Some may need doubling)...............——
7) Unnecessary skylights closed up and insulated——
8) Each check rail window locked tight at night——
9) Radiators in unoccupied spaces shut off 50 per cent or more.........................——
10) Insulate top story ceilings, or under side of roof if piping is in attic..................——

VENTILATION

1) Recirculate all possible air to cut outside air intake to amount actually needed..........——
2) Increase recirculation as temperature drops. Increased infiltration will make up the difference ....——
3) Dead stop outside air intake openings when fans are not running......................——

RECORDS

1) Hourly temperatures inside and out each day——
2) Heating degree days each day and month....——
3) Steam or fuel consumption each day......——
4) Steam or fuel consumption per degree day each day ....——
5) Our steam or fuel consumption per thousand cubic feet per degree day.................——
6) Our neighbors' steam or fuel consumption per thousand cubic feet per degree day.........——

Uncle Sam Wants You to Fight Waste . . . by Heating Efficiently!

"OPEN FOR DISCUSSION"

Communications to the Editor

MOISTURE PICK-UP DATA FOR DRYING BY HOT AIR

The Editor—

Referring to the September HPAC, and in particular to the article on drying of foods, I am very much interested in obtaining information concerning the subject of drying material by means of hot air delivered from a blower. If there are any articles in back issues of the magazine which give information on this subject, I will be glad to secure copies of them.

In particular, information regarding the actual amount of moisture pick-up per cubic foot of air at various temperatures and per cent of humidity would be very welcome. Theoretically, of course, the moisture deficiency of a cubic foot of air can be determined, but the actual amount obtainable in practice is considerably less. If you can furnish me with sources of information concerning this, I shall be greatly obliged.—W. E. H., plant engineer.

Reply (by W. R. Marshall, Jr., technical division, engineering department, E. I. du Pont de Nemours & Co.)—

It is true that in a drying operation in which hot air is used to accomplish the evaporation of moisture, the air will not attain the saturation theoretically possible at the given temperature conditions. The reason for this is that it is neither practical nor economical to dry under conditions which would permit saturation of the drying air. Obviously, one could determine the total minimum volume of air at a given temperature necessary to hold all the water to be evaporated from the substance in question. But this would have no significance from a practical drying viewpoint.

In practice, it is desirable to dry a material at its maximum rate and with the minimum amount of air; i. e., to maintain the ratio of the air circulated per minute to the water evaporated per minute at a minimum. This ratio will decrease as the drying temperature increases and the amount of air exhausted increases. However, here again the economics of the problem enter, since it is not good practice to exhaust hot air which can still be used for drying.

In general, the actual amount of moisture pick-up per cubic foot of air is a function of the following factors:

1) Rate of drying necessary to maintain the desired cycle;
2) Rate of air flow;
3) Air temperature;
4) Dryer construction and insulation.

It is apparent that in the constant-rate period where the drying rate is directly dependent upon the humidity of the drying air, the degree of saturation of this air cannot exceed a certain point, if the desired drying time is to be maintained. Hence, it is necessary to exhaust at a definite humidity a certain percentage of the total air circulated. As the drying enters the falling-rate period, the percentage exhausted can be decreased since the rate of drying is decreasing and hence more air can be recirculated before it reaches a limiting humidity.

The rate of air flow affects the rate at which the air becomes saturated and hence the amount of air to be exhausted.

The air temperature determines the permissible value of the final humidity, which increases with an increase in temperature.

The construction and insulation of the dryer are important in that they determine whether condensation will occur if the temperature of any parts fall below the dew point.

It is not possible to give specific figures of exit humidity unless the data for a specific problem are at hand. However, the figures in the accompanying table may be considered to be generally representative of good practice in hot air dryer operation.

These figures apply to the drying method in which hot air is circulated over a tray of material. They assume that a certain amount of air is recirculated during drying. It is important to realize that if the temperature of any parts of the dryer should fall below the dew point, then condensation will occur which might cause serious corrosion problems, and other complications. In a well insulated dryer, such a situation will not be met.

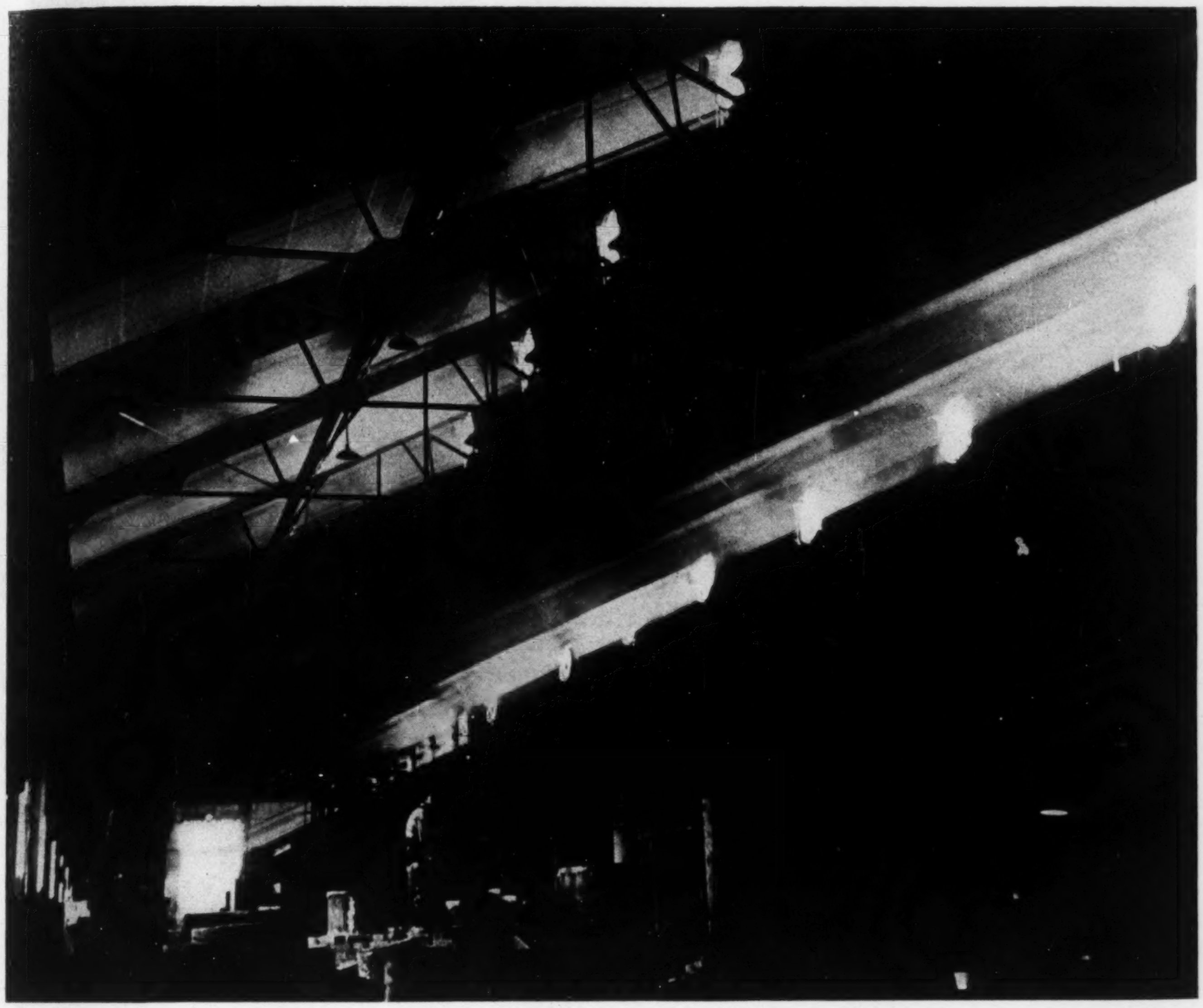
A good illustration of one method of determining the feasibility of recirculating air in a tunnel type dryer for the drying of leather was furnished by O. A. Hougen in a paper on Typical Dryer Calculations, published in Chemical and Metallurgical Engineering, January and March, 1940.

Exhaust Air Temperature, F	Relative Humidity of Exhaust Air, Per Cent	Range of Corresponding Dew Point of Exhaust Air, F	Range of Absolute Humidity of Exhaust Air, Lb Water per Lb Dry Air
120	60—70	100—107	0.043—0.0535
140	60—65	120—125	0.081—0.095
160	50—60	132—140	0.118—0.155
180	35—50	135—150	0.13 —0.21
200	20—35	135—153	0.13 —0.23
220	20—25	147—156	0.19 —0.25
240	12—18	150—160	0.21 —0.29

STOKER PERMITS GAIN

Permits for fuel burning installations in Chicago for the first nine months in 1942 were almost a standoff compared with the same period in 1941, according to a report released by Frank A. Chambers, the city's chief deputy smoke inspector.

A total of 1224 permits was issued in this period in 1942, compared to 1234 for the same months in the previous year. These permits covered all types and sizes of fuel burning equipment, except residential and small apartment buildings. Underfeed stokers showed the only important gain of any of the various types of equipment, with 1023 installations for this period this year as compared with 941 in 1941.



This photo was taken on a typical Monday at 4 p. m., just after eight of the nine furnaces had been fired. The shafts of light are sunlight streaming through the fans

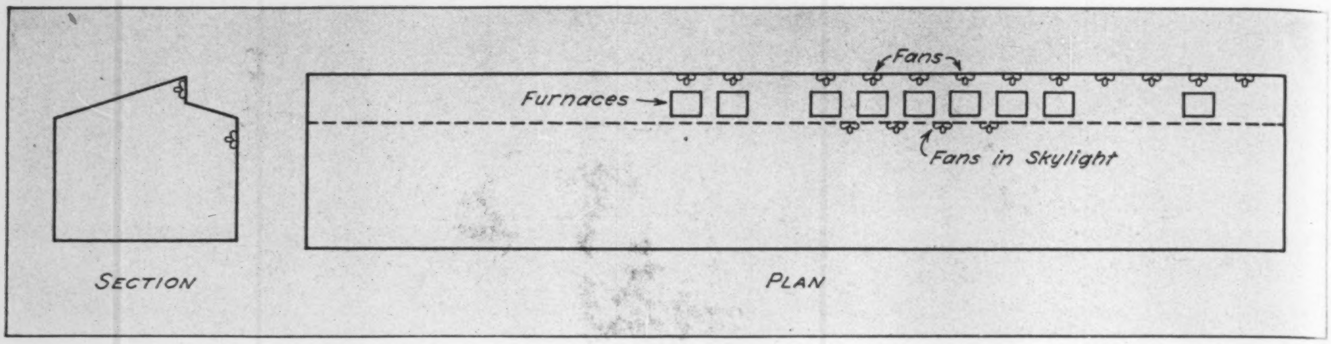
Fume Control in Foundries a Vital Wartime Ventilation Job

How Sixteen Propeller Fans Were Installed to Solve Smoke and Fume Problem in a Busy Foundry . . . . . By Bernard G. Silberstein

SUMMARY — **Adequate ventilation for control of smoke and fumes in foundries is particularly important in wartime because of peak production schedules. The installation of sixteen 36 in. propeller fans described here solved the problem in a busy foundry 320 ft long, 60 ft wide, and 50 ft high. Requiring a total of but 20 hp, and with no ducts or canopies needed, the installation conserved both power and sheet metal. . . . The author is Cincinnati district manager for the Ilg Electric Ventilating Co.**

WITH PEAK production required of foundries during the war's duration, the problem of adequate ventilation for elimination of smoke and dangerous fumes is increasingly acute. The health of foundry employees working 10 to 12 hr a day under trying conditions is of paramount interest to management, for the sickness and possible occupational diseases which result from inadequate ventilation must be avoided.

A Cincinnati foundry had been handicapped for a number of years by the smoke and CO_2 fumes that poured from the gas fired core ovens when green cores were introduced. The production schedule called for firing the furnaces every Monday, and while the firing time was staggered so that the nine ovens were lighted at intervals of approximately every hour, by the time that half of the ovens had been filled with cores, smoke and fumes had filled the foundry to the extent that it was very difficult to see more than 25 ft in any direction. By 4 p.m., when all of the furnaces had been fired, the



Twelve fans were installed along the wall of the foundry directly back of the furnaces and about 15 ft above them

men were withdrawn from the foundry furnace room and it was impossible to see more than several feet because of the severe concentration of smoke. This condition lasted all through the night and it took 24 hr from the initial firing of the first furnace until men could again work in this department.

The building is 320 ft long, 60 ft wide, and 50 ft high. It is of the standard monitor type construction with a series of 48 in. gravity ventilators along the peak of the roof and with steel sash windows for the introduction of air on the opposite side of the room from the furnaces. The gas furnaces are approximately 12 ft wide, 18 ft deep, and 12 ft high. There are nine furnaces along one side of the room.

On very humid days, the smoke problem was at its worst. In addition the radiation and convection of heat from the furnaces themselves was an extra problem, which, while not severe during the winter, caused occasional heat prostration in the summer. An analysis of the gases that were present, particularly on Monday, showed a high percentage of CO_2 in the air.

There were a number of suggestions for meeting this problem. The final solution, engineered by the author, was to install twelve 36 in. propeller fans, running at 1140 rpm and each delivering 15,000 cfm and each with a 1¼ hp self-cooled motor. These fans are along the wall of the foundry directly back of and approximately 15 ft above the furnaces. In addition to these 12 fans, four of the same size fans were installed at the peak of the roof to take care of the surplus fumes that do not go immediately to the lower fans, and to insure satisfactory working conditions for the crane operators. Finally, all of the gravity ventilators were closed off to prevent the air from reversing and entering through them.

The lower level of windows on the opposite wall from the furnaces were opened to insure movement of air over the men at the breathing level. With a ceiling as high as the one involved here, the 16 fans produce an air change approximately every four minutes, which has been found to be sufficient on similar jobs in the past. When a lower ceiling (such as 20 to 25 ft) is encountered, the author has found that an air change approximately every two minutes with a minimum of every three minutes should be used to exhaust fumes rapidly enough. The essential difference between the 50 ft and the 25 ft ceiling is that the higher one offers a reservoir for the fumes. The smoke must be exhausted more rapidly with the lower ceiling.

The results on this job have been extremely satisfactory from the management's standpoint. The accompanying photograph was taken at 4 p.m. on a typical Monday, just after eight of the nine furnaces had been fired; it is apparent that the fumes are being eliminated as fast as they are generated. The interesting shafts of light in the picture are caused by the sunlight streaming through the fans. Needless to add, the production schedule of this foundry has been increased due to this installation.

TESTING FOR METHYL CHLORIDE LEAKS

The Editor—

Installation and service instructions for refrigeration condensing units were published on p. 556 of the September HPAC. These instructions were quoted from a proposed commercial standard for refrigeration units adopted at a manufacturers' conference. In the center column of this quotation, the second paragraph from the top starts, "Test all joints for leaks. Use a good halide torch leak detector for *methyl chloride. . . .*"

A halide torch should *not* be used for testing methyl chloride leaks because of the danger of explosion. In general this dangerous possibility is not known in the refrigeration industry.

Because of their sensitiveness and convenience, halide lamps are widely used for detecting methyl chloride leaks. Ignorance of the explosion hazard is so widespread that even the manufacturers' conference, whose instructions you printed, recommended halide torches for methyl chloride.—W.G.

Contribute to Your Community Fund

Some Practical Pointers on
TEMPERATURE CONTROL

William Goodman Tells of Difficulties He Has Experienced With Various Control Applications and the Remedies

SUMMARY — Control instruments are widely used and have been developed to a high degree of reliability. Nevertheless, there are practical considerations involved in their application which frequently make the difference between a highly successful and a mediocre — or even unsatisfactory—installation. . . . A number of the practical considerations involved in temperature control for several different applications in air conditioning and heating are discussed here. Difficulties with certain widely used types are discussed, and remedies which have proved successful in the author's own experience are suggested. The subject is particularly important because of the need for keeping equipment operating efficiently and without interruption in wartime. . . . Mr. Goodman is consulting engineer, the Trane Co., and a member of HPAC's board of consulting & contributing editors

MANUFACTURERS, THROUGH many years of research and experience, have produced control instruments of sufficient flexibility to permit almost any desired result to be obtained. However, as in other fields of engineering, the instruments must be applied correctly if satisfactory results are to be obtained. In the author's experience, there are certain arrangements of instruments which yield more satisfactory results in actual practice than do others. Practical considerations which determine the success or failure of an installation are sometimes overlooked. A discussion of the practical considerations involved in the use of several types of temperature control installations may be of value to those who design and install air conditioning and heating systems.

Low Limit Thermostats

In hot blast heating systems, a principal thermostat in the room or in the return air duct usually controls the steam admitted to the blast heating coil. In addition to the room thermostat, a low limit thermostat, as shown diagrammatically in Fig. 1, is commonly installed on the discharge of the fan in order to prevent the air from being supplied to the room at too low a temperature. Thus, if the room thermostat is satisfied (that is, the temperature in the room has risen slightly above the setting of the room thermostat) the steam supply would be shut off by the room thermostat if there were no low limit thermostat. However, when the principal thermostat shuts off, the low limit thermostat takes control and prevents the temperature of the air supply to the room from falling too low. The low limit thermostat is usually set to maintain the temperature of the air supply at a minimum somewhere between 60 and 70 deg—usually the latter.

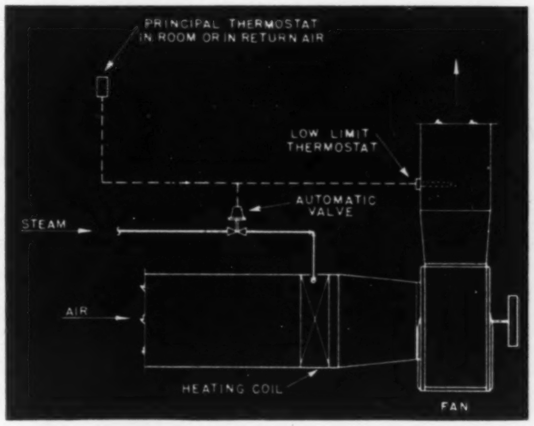


Fig. 1—Diagrammatic illustration of hot blast heating system in which low limit thermostat is used in conjunction with room thermostat

Because of the low limit thermostat, rooms may be overheated. This is especially true in places where large crowds gather, such as ballrooms or theaters. For example, if the room thermostat is satisfied and the low limit thermostat is set at, say, 65 deg, it is quite possible that an air supply at this temperature will be too warm to absorb the heat gain of a room. Even in winter there may be heat gains due to many people, lights, electric motors, and even sunlight. There are many places where for some period of the day a slight amount of cooling is required. The low limit thermostat prevents the air supply from cooling the room. The low limit thermostat, because it will not permit the air supply to enter the room at a low enough temperature, is frequently the cause of overheated rooms.

The low limit thermostat opposes the action of the room thermostat. When the room thermostat tends to shut off the steam supply, the low limit thermostat opens the steam supply. In this way, the action of the room thermostat is defeated. Years ago, when central heating was still a marvel, overheating was not objectionable; in fact, it was frequently taken as a sign of a well designed system with adequate capacity. The demands of today, however, are more exacting. Overheating is as much to be avoided as undercooling.

It has been the author's experience that the use of low limit thermostats frequently interferes with the proper control of temperature in the room and that it is virtually impossible to control the room temperature satisfactorily if a low limit thermostat set above 60 deg is used. A low limit thermostat may be satisfactory providing it is set at approximately 45 or 50 deg. However, if the low limit thermostat is to be set at this temperature, there is really no need for installing one. In cold weather the room thermostat itself will usually call for sufficient heat so that the air supply will not drop much lower than 50 deg, even in crowded auditoriums.

With faulty introduction of air into a room, drafts are likely to be less objectionable with warm air than with cold air. In the past, when little was known about correct

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methods of supplying air to rooms, and grilles and registers of proper design for avoiding drafts were not available, there was no way of supplying low temperature air to a room without causing uncomfortable drafts. The design and location of grilles was dictated mainly by architectural considerations rather than considerations of draftless air introduction and distribution. Low limit thermostats, which kept the temperature of the air supply close to the temperature of the room air, were the only means available of minimizing the effects of drafts caused by faulty introduction of the air supply.

With the many types of correctly designed air supply inlets available today, however, and with the knowledge available for proper location of these inlets in rooms, there would seem to be little need for limiting the temperature of the air supply when the principal room thermostat is satisfied. Drafts should be prevented by the proper introduction of air into a room through properly designed air inlets rather than by limiting the temperature of the air supply during periods when the room thermostat is satisfied. Air at low temperatures can be successfully introduced into a room without drafts if the air inlets are properly selected and placed.

Multiple Thermostats

Frequently, two separate pieces of equipment must be controlled from the temperature in the same room. For example, in an air conditioning system two separate compressors may occasionally be used. The compressors may be set to start, say, 2 deg apart. Thus, if the temperature should rise to 80 deg, the first compressor will start. If, in spite of the running of the first compressor, the temperature in the room should continue to rise to, say, 82 deg, the second compressor would start.

There are many other types of installations in which two separate pieces of equipment may be used to cool or heat a single chamber. In such a case, two separate thermostats set about 2 deg apart are frequently installed. Although two separate thermostats can be set to operate in this manner, the results will not always be satisfactory.

In the first place, there is the difficulty of adjusting two ordinary commercial thermostats to operate consistently a few degrees apart over a long period of time. The temperature scales on ordinary, uncalibrated thermostats — the type commonly used—cannot be depended upon in setting two thermostats to synchronize properly in starting and stopping equipment.

In the second place, no one knows exactly where the thermostat should be set to suit the vagaries of the human system. One day a temperature of 78 deg may feel comfortable to the occupants of a room, while on another day they may require 74 deg. It is impossible for the ordinary, untrained person to adjust the two separate thermostats so that they will continue to function as the designing engineer intended them. If the occupant of the room changes the setting of one thermostat, he will rarely be able to change the setting of the other one to operate on, say, the 2 deg difference which the designer intended. Even trained operating engineers have difficulty in changing the temperature setting of the two ordinary instruments so that they will synchronize properly. The least that can be said of an arrangement using two thermostats is that it is highly inconvenient when changes in setting must be made.

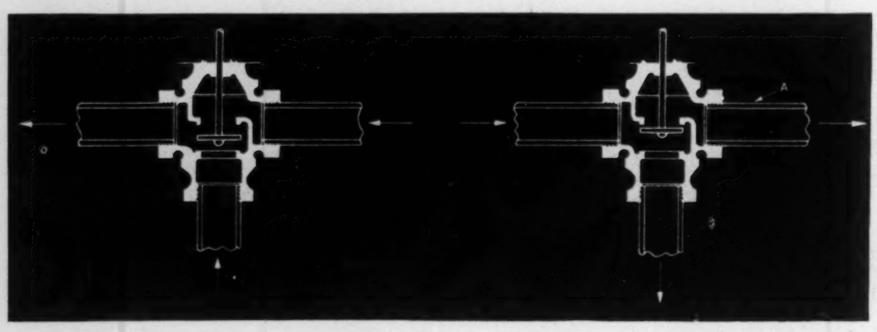
For these reasons, it has been the author's experience that two separate thermostats installed in this manner give unsatisfactory results. It is better to use a single, two step thermostat or else a modulating thermostat with a step controller. In this way, regardless of where the user sets the thermostat, the setting of the second contact will always be made automatically to maintain the 2 deg (or other) difference intended by the designer.

Three Way Valves

Three way mixing valves are frequently used not only in heating and air conditioning, but also in industrial control processes. The ordinary three way mixing valve works best with two supply connections and one outlet connection. In other words, the liquid flows in through two inlet connections as shown in Fig. 2 and out through one outlet connection. If an ordinary three way valve is connected in the reverse manner —that is, with two outlet connections and one inlet connection as shown in Fig. 3—the valve will not operate properly when the disc is very close to either seat. For example, with the disc in the position shown in Fig. 3 very little liquid will be passing through the upper seat. As a result, there will be little or no pressure in the pipe marked "A"—especially if pipe "A" is vented to the atmosphere. In this case the disc will become unbalanced because of the pressure under the disc and the lack of pressure above the disc. Consequently, regardless of the control means, the pressure will tend to slap the disc shut against the upper seat.

Movement of the disc due to unbalanced liquid pressure occurs frequently when three way valves are connected with two outlets and one inlet. Whenever the disc is in either extreme position, poor control results—often to the point where complaints may be voiced. To avoid this difficulty, three way valves should always be so installed in the pipe line that there will be two inlet connections and one outlet connection, as in Fig. 2. In this case, the

Fig. 2 (left)—Correctly installed three way mixing valve with *two* inlets and *one* outlet
Fig. 3 (right)—Incorrectly installed three way mixing valve with *one* inlet and *two* outlets



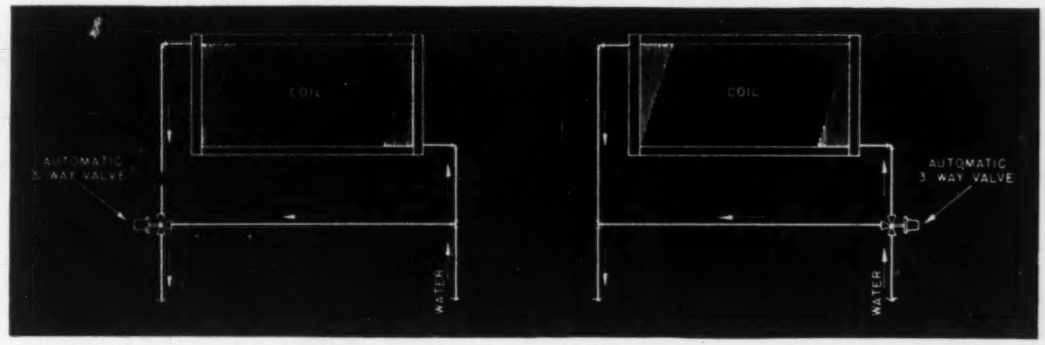


Fig. 4 (left) — Correctly installed three way mixing valve on water coil. Fig. 5 (right)—Incorrectly installed three way mixing valve on water coil

pressure above and below the disc will always be balanced and the movement of the disc will be controlled by the temperature control element. For air heated or cooled by coils inside of which water is flowing, the three way valve should be installed on the *outlet* side of the coil as shown in Fig. 4, instead of on the *inlet* side as shown in Fig. 5.

Preventing Freeze-Ups

When heating an outside air supply, two heating coils are often installed, as in Fig. 6, in order to prevent freeze-ups.

The steam supplied to the first coil is controlled by means of an on and off valve and not by a throttling valve. The thermostat controlling this on and off valve is situated in the stream of outside air. Generally the thermostat is set to turn steam into the first coil whenever the outside temperature falls below about 34 deg. In this way, the possibility is eliminated of admitting air at a temperature lower than 32 deg to the second coil, which is controlled by a throttling valve.

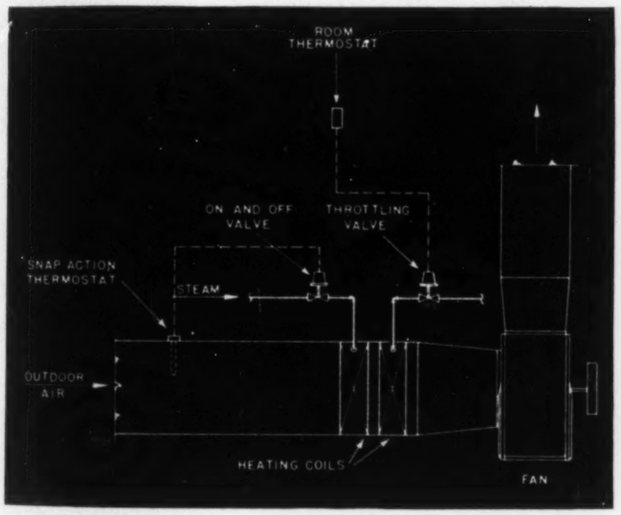


Fig. 6—Diagrammatic illustration of blast heating system with tempering coil for outdoor air to prevent freeze-ups

A throttling valve is installed on the steam supply to the second coil. A thermostat in the room or in the return air regulates the steam flow through this valve in accordance with the room temperature.

There is always great danger of a freeze-up if air at a temperature below 32 deg is passed over a coil controlled by a throttling valve. For example, on light loads, steam might be admitted only to the upper portion of the coil by the throttling valve. If the air were not preheated, the cold air passing over the lower portion of the coil would cool the condensate and eventually freeze it, causing damage to the coil. By preheating the air with the first coil whenever the outdoor temperature is lower than 32 deg, the danger of freezing in the coil controlled by the throttling valve is eliminated.

The difficulty with using two coils in series—and with one of the coils controlled by an outside thermostat—is that the first coil is sure to be on when the outdoor temperature is about 34 deg or lower. Most commercial coils available today will heat air entering at about 30 deg to a final temperature of approximately 60 deg. In many cases, particularly where the internal heat gains are high, an air supply at 60 deg may cause overheating in the room to which the air is supplied. It has been the author's experience that such overheating frequently occurs in spaces where many people are gathered. The overheating of many such places in mild winter weather can be traced directly to the fact that the preheating coil used for prevention of freeze-ups supplies air that is too warm to maintain the required room temperature.

One way of eliminating this difficulty is to use only a single coil controlled by face and bypass dampers. A room thermostat controls the damper motor, which in turn moves the face and bypass dampers as shown in Fig. 7. In addition, the steam is controlled by an on and off valve which is always open when the face damper across the coil is open. The steam valve stops the supply of steam whenever the dampers across the face of the coil are closed and the bypass damper is open. The control should be so arranged that before the room thermostat can open the damper across the face of the coil, the steam valve is wide open and steam pressure is available inside the coil. The on and off steam valve is needed because there will always be a certain amount of air leakage through commercial dampers. In addition, because dampers are usually installed on only one face of the coil, there is always considerable impingement of air on the exposed face of the coil. Consequently, if the coil were kept hot there would be danger of overheating. The steam valve, by shutting off the steam when the damper closes, prevents overheating due to leakage and impingement.

It would be desirable to eliminate the on and off steam valve and leave the steam in the coil at all times. This can be accomplished by using two sets of dampers on the coil, one set on each face as shown in Fig. 7. The blades of the damper should be constructed of a double strip of sheet metal with a sheet of asbestos or air cell insulation sandwiched in between the two strips of metal as shown in Fig. 8. In addition, the interlocking edges of the dampers should be lined with felt. With double insulated dampers built as described, and with the edges lined with felt, overheating of the air due to leakage and impingement can practically be eliminated even though the coil is kept full of steam at all times. This is highly desirable as it eliminates the danger of a freeze-up at any time.

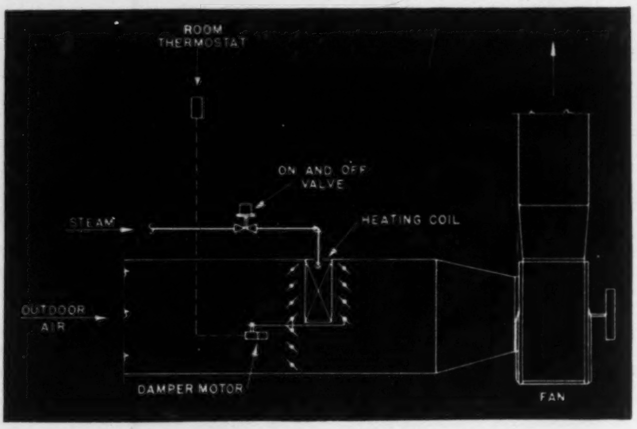


Fig. 7—Diagrammatic illustration of blast heating system with single coil and face and bypass dampers

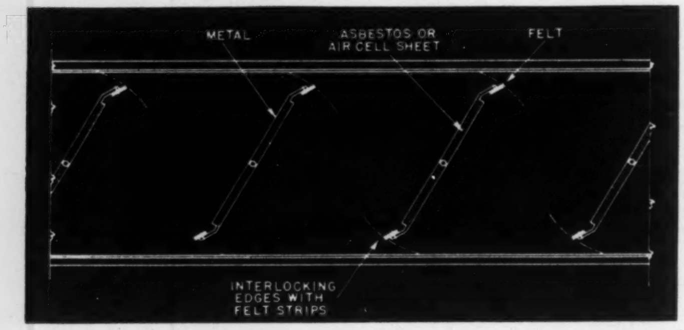

METAL
ASBESTOS OR AIR CELL SHEET
FELT
INTERLOCKING EDGES WITH FELT STRIPS

Fig. 8—Suggested construction for face dampers of Fig. 7

By using face and bypass dampers in this manner, the room temperature can be accurately controlled at all times without the danger of overheating in mild weather which usually occurs when using two separate coils.

Another method of avoiding the use of two separate coils is by using coils made with internal steam distributing tubes. By means of the internal distributing tubes, steam is distributed equally to all parts of the coil even though a throttling valve is used to control the steam supplied. The throttling valve is controlled by means of a thermostat in the room or in the return air. However, in spaces such as ballrooms and theaters, the heat gain due to the people may at times almost entirely offset the heat loss from the building even though the outdoor temperature is around 30 deg. At such times the steam load on the coil may be very small and the throttling valve may assume an almost closed position. When the throttling valve is nearly closed, the pressure in the distributing tubes of the coil may fall to a low point and distribution of the steam over the inside surfaces of the tubes may be impaired.

Still another method sometimes employed utilizes the mixing of cold outdoor air with warm return air so that the temperature of the resulting mixture entering the heating coil will be above 32 deg. The mixture is so proportioned that the temperature of the air entering the coil is usually about 40 or 45 deg. This plan would be excellent if it were possible to thoroughly mix the outdoor air with the return air ahead of the heating coil. Unfortunately, space restrictions are commonly such that thorough mixing seldom occurs. It has been the author's experience that stratification invariably takes place, and air entering some portions of the coil will be at practically the same temperature as the outdoor air. Because of stratification, therefore, the danger of freezing the coil is great even when return air is mixed with the outdoor air ahead of the coils. Unless some means is provided for thoroughly mixing the streams of outdoor and return air so that the mixture entering the coil is at a uniform temperature at all points across the face of the coil, this method is not a dependable one to use.

Starting and Stopping Fans

In small heating and cooling systems the room temperature is sometimes controlled by means of a thermostat which starts and stops the fan. This method, in the author's experience, frequently is unsatisfactory—particularly during those portions of the heating season when the outside temperature is low. Periods of crisp, cold weather may be invigorating when one is outdoors, but it is during such periods that the lack of satisfactory heating is most keenly felt. When the fan stops, stratification of the air takes place almost immediately. Anyone who has ever sat in a room heated by warm air knows the chilly feeling that is experienced when the fan stops. Such rooms are comfortable only when the fan is running.

To eliminate this difficulty, it is better to use a two speed motor so that when the room thermostat is satisfied, the fan, instead of stopping completely, will run at half speed. In this way stratification will be prevented and a more uniform temperature will be maintained in the various rooms supplied by the fan. In rooms in which the fan is run at either high or low speed during cold weather, the occupants are not likely to experience the difference in feeling which occurs due to the starting and stopping of the fan.

There are many systems for controlling the heating means and the fan together. Usually the room thermostat starts and stops the oil burner, stoker, or gas burner. The circulating fan is controlled by a thermostat in the bonnet of the furnace. When using a two speed motor, it is desirable to use a two step bonnet thermostat. In cold weather, the fan would cycle between high speed and low speed in accordance with the demands for heat in the room.

During mild weather it is likely that the fan would never get up to high speed because it would be able to provide all of the heat needed when running only at low speed. Therefore, the thermostat would keep the fan off the major portion of the time and at low speed during the little time that heat must be provided. When the temperature is high outdoors, the starting and stopping of the fan cannot be detected by the occupants of the room. Furthermore, because the fan runs at low speed when it is on, much more uniform and satisfactory heating is likely to be obtained. In this case the fan running at low speed will be on a longer period of time than if it were running at high speed.

The two speed fan driven by a motor with two separate windings, even though more expensive, is superior to the usual on and off arrangement and usually provides more uniform temperature in all kinds of weather, with resulting greater satisfaction with the heating plant.

Outdoor Air Damper

Frequently in air conditioning systems a damper is provided on the outdoor air inlet; this inlet is usually of such size that the entire air supply can be taken from the outdoors. In early spring and late fall, the outdoor air damper is usually kept wide open and the return air damper is closed. However, in either extremely warm weather or extremely cold weather, it is customary to use mainly return air, taking only a small percentage of outdoor air. At such times the outdoor air damper must be almost wholly closed. Usually a hand operated switch is provided to set the outdoor air damper in any one of several positions—quarter open, half open, etc.

The position of the blades is adjusted at the time that the temperature control is installed to correspond to the various markings on the hand switch. However, it is virtually impossible to accurately set the angle of the blades to obtain any given amount of outdoor air. Furthermore, the setting of the damper blades may change over a period of time due to corrosion or dirt in the pivots.

Usually two or three separate damper motors and dampers installed on the outdoor air intake will be more satisfactory. The dampers are either wide open or wholly closed. For example, one damper may cover 20 per cent of the intake, and the other two dampers each cover 40 per cent. In this way, it is possible in extreme weather to open only 20 per cent of the outdoor air intake and to know that when the hand operated switch is operated that a definite area of the intake is open. This method usually gives much better control of the outdoor air than the one in which an attempt is made to set a single large damper in various positions.

* * *

It is not easy to translate personal experiences with equipment difficulties into general rules that can be used by others as infallible guides. Even though each man may arrive at the same cause for the difficulty, each may provide a different and satisfactory solution for eliminating the difficulty. There is frequently more than one solution for any given problem. No claim is made that the solutions given here are the best; they are only the solutions which in my experience have worked well. They may, therefore, be of help to others.

13,200 TONS OF AIR CONDITIONING FOR NEW WAR DEPARTMENT BUILDING

Some time about mid November, the War Department will have a new "Army of Virginia." It will move into the world's largest office building—a building covering large acreage and having a perimeter of approximately one mile.

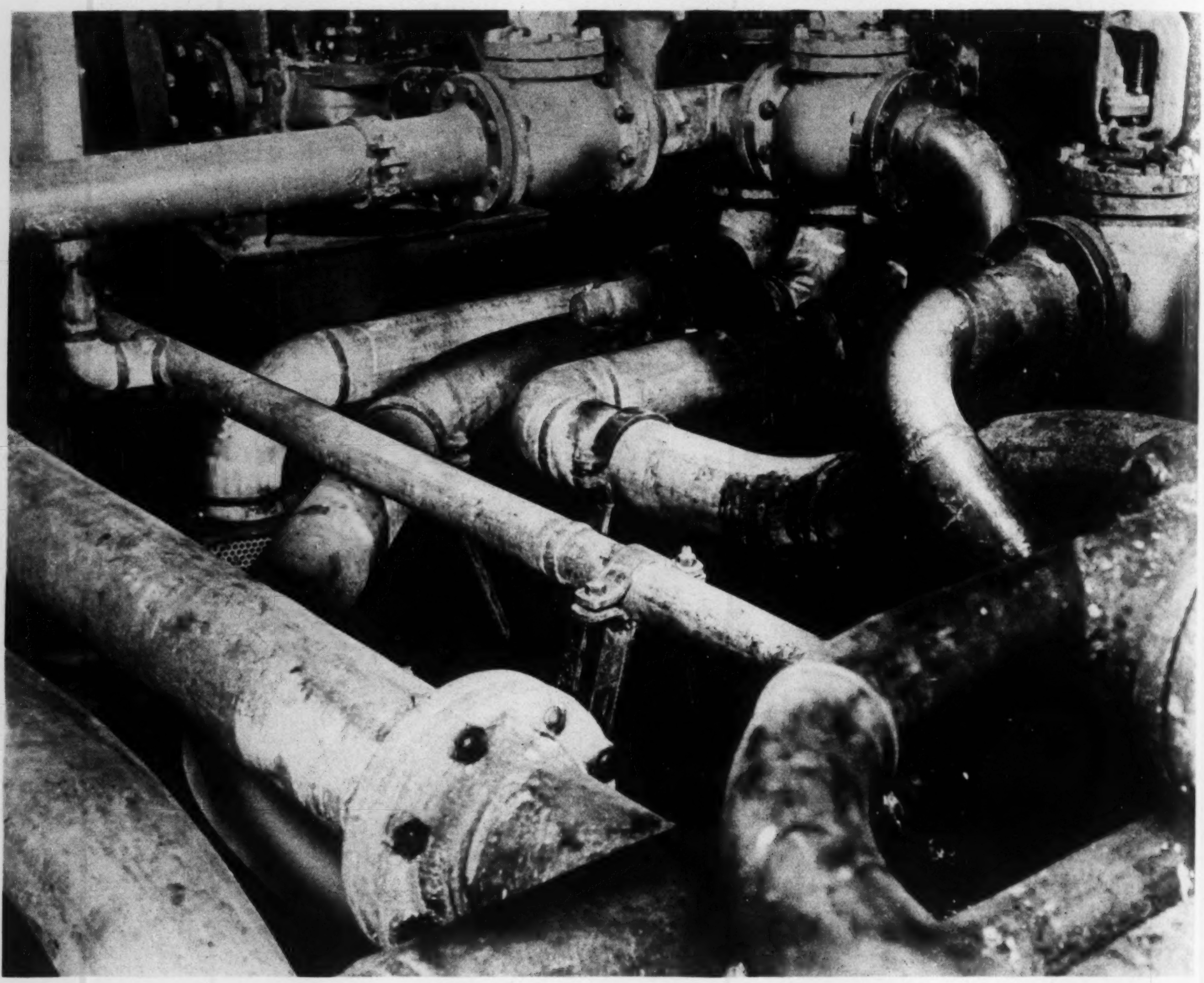
The building, of pentagonal shape with specially designed communicating corridors to handle the 35,000 employees or over, is also completely air conditioned, to keep manpower efficiency at its peak in the humid weather on the banks of the Potomac.

To get some idea of the size of this air conditioned building, just start with the Yale Bowl or some comparable stadium. That is about equal in size to the court yard of the Army's new home made necessary by the size of America's new Army. The Empire State building is taller than this mammoth beside the Potomac. The Merchandise Mart may look larger. These and others may cost more because of site and trim. But the new Pentagon building tops them all in floor space, although it is only five stories high. The Army figures it will save the taxpayers $4,000,000 a year now being paid in 17 or more buildings scattered over Washington, with a resultant lack of coordination.

The building is five-sided so that it can be approached from five directions—and each approach will give a full-faced view of the building. Actually it is five buildings in one—four nested within one outer building in an ingenious, space saving design that gives a maximum of light and convenience. On the first and second floors the space is continuous, with the exception of the interior road. The upper three stories are arranged in a series of rings separated by light walls.

Twelve centrifugal refrigeration machines were supplied to provide a cooling capacity of 13,200 tons. There will be 7800 window air conditioning units installed throughout the building to provide a total of 3,900,000 cfm of conditioned air. Condensing water from the Potomac will circulate at the rate of 46,600 gpm. This water will be used and then turned back to the river. In the cooler of the centrifugal refrigerating equipment, 25,900 gpm of water will be chilled for cooling the air that goes to the rooms. This water is used over and over again, flowing from the central plant through the air cooling system and back again without loss of water.

The design and construction of this 142 acre building is under the direction of Lt. Col. Clarence Renshaw. G. E. Bergstrom and D. J. Witmer are chief architects and Charles S. Leopold is mechanical engineer. The general contractor is a joint organization of John McShain, Doyle & Russell, and Wise Contracting Co., Inc.; the mechanical subcontractor for the refrigeration plant, Carlson Brothers; for the building conditioning, Baker, Smith Inc. and Mehring & Hanson. — Margaret Ingels, engineering editor, Carrier Corp.



Advantages of Welded Piping for Speeding Shipbuilding

W. H. Girdler, Jr., Tells How Welding Fittings and Piping Assemblies Can Simplify Prefabrication and Reduce Time for Building Vital Cargo Ships

SUMMARY—As indicated in the picture at the top of this page, space is at a premium aboard ship—there is never enough of it. Mr. Girdler, of Tube-Turns, Inc., tells here how welded piping systems can simplify ship design and construction, and thus speed our vital shipbuilding program. The sketch on the next page shows the use of welding fittings in a small section of an auxiliary exhaust piping system in the vicinity of a couple of pumps, the dashed circles illustrating typical welded constructions used to advantage

TODAY, MORE than ever before in the history of shipbuilding, the industry is facing grave problems. Uppermost in the minds of all shipbuilders, all allied and dependent shipping industries, and the armed forces, are these questions: How soon can we build the necessary ships? What is the minimum time from laying the keel until the ship is ready to sail with cargo?

Prefabricated and preassembled sections of piping systems are the answers to some of today's shipbuilding problems. In order to speed construction, save materials and space and weight, welded piping construction must be used extensively. It has been well established that a weld is normally stronger than the parent metal, so there is no difficulty from the strength viewpoint.

On every vessel constructed, space in the engine room for passageway is at a premium and the more compact the piping around various units, the more space available for passageway. Welding saves space.

By using welded fittings (welding ells, tees, and reducers, etc.), the number of possible joint leaks is reduced. This conserves steel, too. The welding together of several fittings also permits an easy method of preassembling quite a sizeable run or section of piping around a unit—thereby simplifying testing of it before installation aboard ship.

Nearly all piping systems may be made much lighter on the whole by the use of welding fittings, for the wall thickness of these fittings may be the same thickness as the pipe. In systems where space is very limited and the pipe is of such size that the proper radius cannot be obtained in bending such pipe, it is usually possible to substitute welding fittings to obtain the desired results.

The inside walls of welded fittings are smooth and clean and thus restrict very little the flow of fluids. If fittings and pipe are properly welded together and cleaned well on the inside, restrictions to flow are at a minimum.

It is possible to eliminate even more joint leaks by using welded end connection valves in systems in some cases; this is a point that should be carefully studied. In piping systems that require covering for reason of heat and temperature control, the welded fittings and welded pipe joints make for easy and neat covering, thus cutting down on the necessary man-hours of labor.

The accompanying sketch illustrates the use of welding fittings in a small section of an auxiliary exhaust piping system in the vicinity of a couple of pumps aboard ship. The layout has the following advantages over the original design:

Saving of 11¾ in. overall space, which was greatly needed in the passageway.

Elimination of about 14 possible leak joints.

Total saving of 14 gaskets.

Easier covering operation (this piping is normally covered), and assembly time. Man-hours required to do the job are thereby reduced.

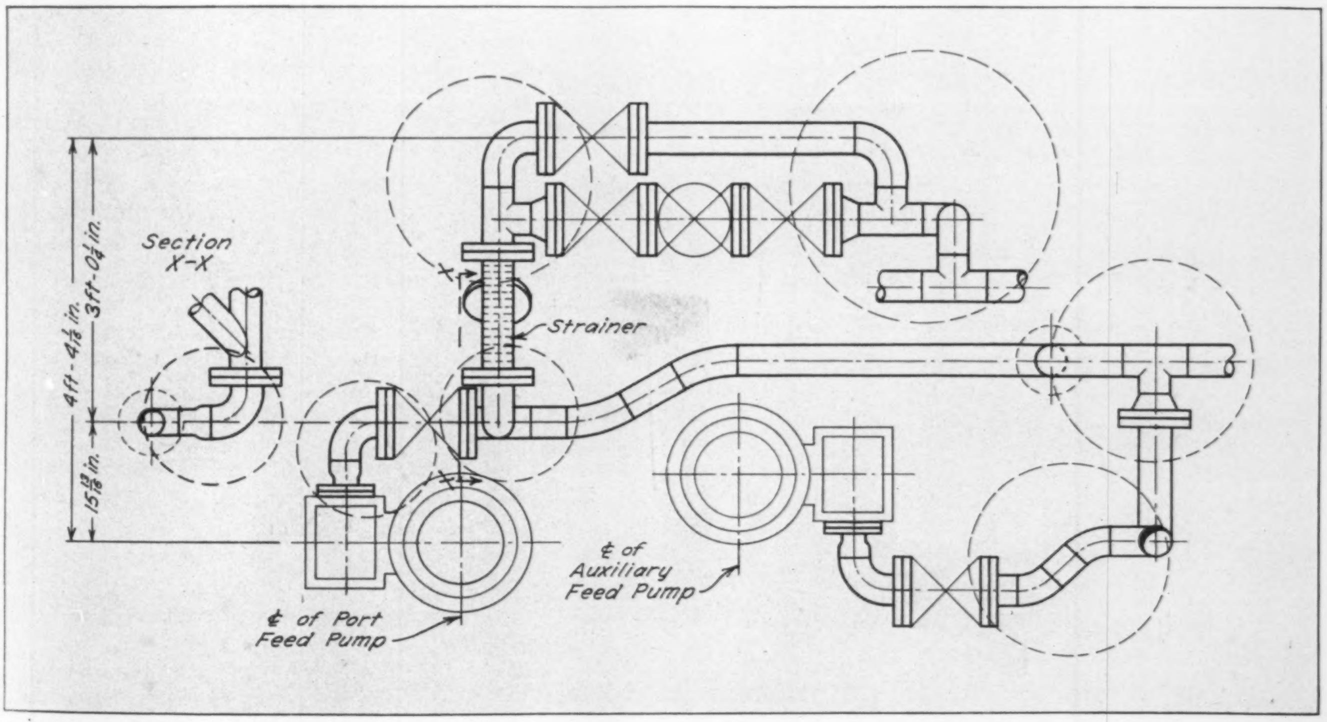
Saving of approximately 600 lb of weight by using welding fittings.

Simple preassembling of sections between flanges, and easy shop testing.

Enclosed in the dash line circles on the sketch are typical welded constructions used to advantage. It should be borne in mind that considerable time, expense, and weight can further be saved by using welded end connections on valves.

Systems can nearly all be simplified through the use of welded construction, with proper foresight on the part of engineers and designers. The proper placing of necessary "breaking joints" is merely a question of knowing ship piping design, which is acquired through experience in this work.

Shipbuilding records being set at the present time are largely due to modern construction practices, such as prefabrication and preassembly, and a very great amount of welded construction—both of hull work and welding fittings in piping systems. For instance, when passing pipe lines through small spaced cofferdams, it was very often necessary to make longer runs of pipe and ugly and difficult hook-ups in order to allow necessary clearances. Larger holes than desired were very often cut to facilitate this procedure; cumbersome and weighty bulkhead fittings were often needed. Today, however, many of these problems have been eliminated, due to the development of welding fittings. Special drilling for castings turned at an angle (to allow for better arrangements) is not necessary with welding fittings; slip on welding flanges, drilled standard and turned to match the companion flange, with the fitting at any desired angle, and with the flange welded to the fitting, can be used..



WAR PROBLEMS FORUM

TWO HUNDRED SALVAGE WARDENS HUNT SCRAP

Two hundred salvage wardens patrol workshops of the Westinghouse Electric and Mfg. Co. to hunt out scrap and promote full use of vital materials. Modeled after the civilian defense units which have been set up in every American community, the salvage warden plan will operate continuously and will extend to other plants of the company after being tested at one of the works. On each scrap hunter's left arm is a blue, red and yellow brassard, similar to the Office of Civilian Defense insignia worn by air raid wardens. The arm band bears the legend "salvage warden" and the V for victory sign.

Four workmen in each section were appointed zone wardens, each man serving a full week every month. They are under the direction of district wardens, who meet weekly with the chief warden, thus establishing a chain of command from the top down to the last zone warden. At the weekly meetings, the district leaders report the findings of their men, review results, and plan future action.

First aim of the wardens is to reduce the amount of scrap. They are under orders to make every effort to find a use in production for every bit of material. If a steel plate, for example, is not suitable on the job for which it was ordered, the warden checks other departments to see if they can use it. The plate goes to the scrap pile only when it can serve no other wartime purpose.

Next, they hunt for dormant scrap—old machinery that cannot be repaired, useless materials stored in out of the way corners, broken or obsolete tools in drawers and on benches.

Ten million pounds of scrap were collected in the last six months at one plant and the wardens expect to dig out several more million pounds before the year ends.

To prevent mixing of different kinds of metals, a sharp watch is kept on scrap bins placed beside cutting machines. If copper chips are contaminated with steel shavings, the resulting metal cannot be used again where pure copper is required.

Every piece of steel, copper, aluminum and rubber, every idle machine and every operation will be checked by these men, and it is estimated that a 25 per cent increase in savings of materials will result.

Thirty tons of idle steel, copper and other metals will be taken from an abandoned service plant which is being dismantled. More than 8000 obsolete dies and jigs, made of high grade steel, were released from storage and added to the scrap pile.

Get Your Scrap in the Fight!

Salvage for Victory

BUREAU OF MINES STUDYING STOCKPILING OF COAL

To assist commercial coal users in minimizing storage losses, the division of solid fuels utilization for war of the Bureau of Mines is organizing a group of experts to advise industries on best methods of stockpiling coal to avoid spontaneous heating and degradation of the stored product, Dr. R. R. Sayers, director of the bureau, has announced.

Industrial users of coal have established or are planning to accumulate larger reserves to avoid possible transportation difficulties, Dr. Sayers informed Secretary of the Interior Harold L. Ickes, and it is extremely important that damage to stockpiles be controlled so that war production will not be impaired. Coal storage problems have increased because larger amounts are being stored and many new consumers now are stockpiling coal for the first time.

The Bureau of Mines is making a survey of the action of coal in storage under various conditions in all parts of the United States. To hasten the date when the bureau's new service will be fully effective, Dr. Sayers requested that persons encountering difficulties in the storage of coal in commercial quantities immediately inform the Bureau of Mines. Simultaneously, the bureau is sending questionnaires to commercial users of coal throughout the nation to obtain the case histories of various coals in storage. Data obtained from these questionnaires will be assembled in Washington, D. C., so that the information may be summarized and made available to those seeking solutions to storage problems.

J. F. Barkley, chief of the division of solid fuels utilization for war, said that if proper methods are followed in the storage of coal, the loss from spontaneous heating will be negligible. In spontaneous heating, oxygen from the air unites with coal, and if a certain temperature is reached, the coal begins to burn. Tests by the Bureau of Mines show that the heating value of most coals suffers but little from storage. Higher rank coals lose a maximum of about 1.2 per cent during the first year and 2.1 per cent in two years. The lower rank coals may lose as much as 2 to 3 per cent in the first year and as much as 5 per cent in three years, experiments have shown.

IF IT HASN'T BEEN OR WON'T BE USED FOR THREE MONTHS—SCRAP IT!

To move every pound of critically needed waste materials into the flow for the manufacture of implements of war is the overall objective of the salvage branch of the conservation division of the War Production Board. To make every man, woman, and child in America conscious of this need, and to secure the active cooperation of every American to "get in all the scrap—*now,*" is equally important. It has been determined that 17,000,000 net tons of purchased iron and steel scrap is wanted in the second half of 1942 to give the iron and steel industry sufficient inventories to prevent shutdowns when the cold weather sets in.

The WPB industrial salvage section is charged with the vital responsibility of educating industry to salvage all critical waste materials, to speed up the return of these materials into the war production stream, and to help industry to use established channels of disposal.

The present immediate objective of the section is to urge and assist the executive management of every industrial establishment in the country to locate, classify, and move into war production channels all dormant and production scrap in their possession as speedily as possible. "Dormant" scrap is defined as obsolete machinery, tools, equipment, dies, jigs, fixtures, etc., which are incapable of current or future use in the war production effort because they are broken, worn out, irreparable, dismantled, or in need of unavailable parts necessary to practical re-employment.

Hamilton W. Wright, chief of the industrial salvage section, says to industry: "Worn out, unusable material, machinery, and equipment must be reborn in America's furnaces, smelters, and refineries. Only by this method can we obtain sufficient raw materials required to produce urgently needed armament for our fighting forces."

Representatives of the industrial salvage section in all parts of the country are urging industrial firms to appoint an executive as salvage director, put into motion a practical, continuing salvage campaign, and to adopt the policy: "If it hasn't been used for three months, and if someone can't prove that it's going to be used in the next three—find a use for it or scrap it."—Paul C. Cabot, deputy chief, conservation division, War Production Board.

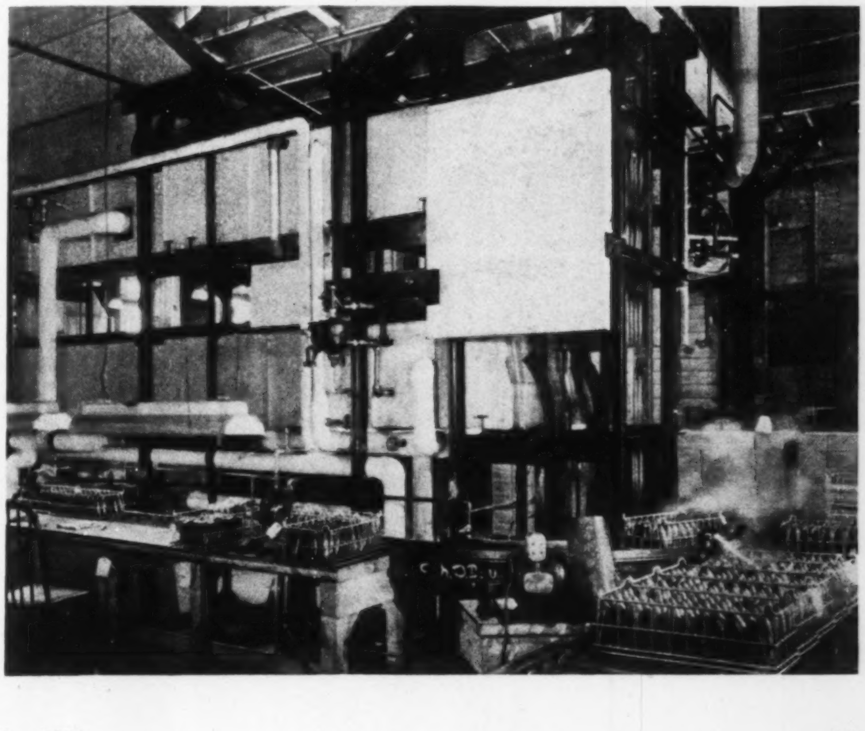
REFRIGERATION UNIT FOR SHOCK TESTING AT ORDNANCE PLANT

Air conditioning and commercial refrigeration dealers, by using their peacetime engineering experience with resourcefulness and ingenuity, are making important contributions to the war effort. One interesting example is the shock testing equipment designed and built by the Conditioned Air Equipment Co. for an ordnance plant.

The purpose of the equipment is to shock test 37 mm shot before inspection. The shot is armor piercing and if any strain, stresses, or defects are in the shot, the shock test cracks them open and the shot is rejected. The problem was to build this equipment to take up the least possible floor space and to tie it in with a streamlined production line. The unit takes up only 26 ft by 26 ft of floor space and requires only the part time services of one girl to load and unload.

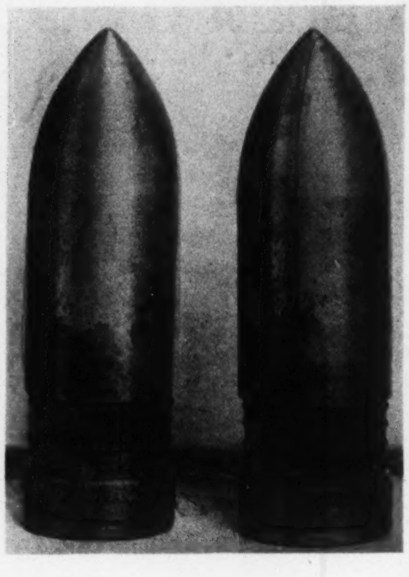
The unit consists essentially of a conveyor system leading in and out of a series of insulated tanks for water at 40 F and 212 F. The shot is brought from room temperature to 40 F (specifications call for cooling below 60 F), in one water bath, is then plunged into a second bath with water at 212 F. In a third bath it is brought down to 40 F again. From here it goes through a soluble oil bath to prevent corrosion.

Refrigeration for the unit is provided by a 5 hp condensing unit connected to a cooler. The condensing unit, water cooling unit, and a circulating pump are mounted be-



View of the shot testing equipment in an ordnance plant low the tanks. A standard gas fired boiler supplies heat for the 212 deg tank.

An interesting sidelight is a drinking water coil in the bottom of the cold water tanks. This is connected with a bubbler system to provide cold drinking water for employees. Because of health considerations and high room temperatures in the plant, it was necessary to place a mixing valve in the line so that the temperature of the water at the bubblers may be maintained at any desired level.—R. C. Lindblom, General Electric Co.



Samples of shot after having passed through cold and hot baths in shock testing unit. Shot at left is perfect. Shot at right cracked because of defects

HOW TO MAKE FULL USE OF ALL MOTOR HORSEPOWER

In a drive to get more war production out of electric motors and at the same time save up to 40 per cent of the copper, steel, and aluminum used in their manufacture, we recently began a "calling all horsepower" campaign. The War Production Board has asked motor users to load each unit to the limit.

To make full use of all motor horsepower, these recommendations should be heeded:

1) Motors open to the air should be overloaded 25 per cent above their nameplate rating unless the surrounding temperature is exceptionally high. Enclosed motors can be overloaded 10 per cent and direct current open motors 15 per cent under the same conditions.

2) Most plants using small motors have 220 or 440 volt power supplies. WPB has asked that 440 volts be used because this reduces the amount of copper required in leads and in the control apparatus.

3) Use standard sealed sleeve bearing motors unless ball bearing motors are absolutely essential. Ball bearings are urgently needed in other war machinery.

4) Use motors of highest practical standard speed and avoid use of multispeed motors. Faster motors are smaller and require less critical materials to build.

5) Use alternating current motors in preference to direct current types wherever possible. This saves substantial amounts of copper and helps alleviate a shortage of direct current motor manufacturing capacity.

6) Match control and circuit protective equipment to the motor on the basis of nameplate horsepower rating, thus getting the maximum use from control equipment.—A. C. Streamer, vice-president, Westinghouse Electric and Mfg. Co.

BUY WAR BONDS
AND STAMPS

PURCHASING OFFICERS NEEDED FOR CIVILIAN WAR SERVICE

The enormous expansion of government establishments connected with the war effort has vastly increased the volume and importance of the work of government purchasing officers.

Purchasing officers are now being sought for positions which pay from $2000 to $4600 a year. They will prepare specifications for the purchase of government supplies; prepare invitations to bid; and maintain current information with respect to market trends, fluctuations, sources of supply, and laws and regulations pertaining to federal procurement.

The civil service examination announced for filling these positions calls for persons who have had from two to six years of responsible experience as purchasing or procurement officers handling large lots of materials of considerable variety. This experience may have been acquired with a large railroad, or other public utility, a large industrial or commercial establishment, a branch of the federal government, or with the government of a state or large municipality.

There are no age limits for this examination. No written tests are required. Applications will be accepted until the needs of the service have been met. Announcements and application forms may be obtained at any first or second class post office or from the Civil Service Commission, Washington, D. C.

M-219 RESTRICTS SILICA GEL

General preference order M-219, issued recently by the WPB, limited as of October 1 use and delivery of silica gel except as specifically authorized by the director general for operations upon application pursuant to the order.

Applicants must specify, if consumers, the use of the material, which includes packaging implements of war, dehumidification of air space in ships, dehydration of foods, preparation of pharmaceuticals, generation of oxygen, deleading of gasoline, gas mask manufacture, preparation of blood plasma, refrigeration, air conditioning, and "other."

THE DRYING OF FOODS

War Creates Tremendous Demand for Dried Foods to Save Shipping and Storage Space, Prevent Spoilage ...... W. R. Marshall, Jr., Describes the Drying Fundamentals Involved

In the design of chemical engineering equipment, design calculations are based, wherever possible, on fundamental theory. In many instances, however, these calculations must be based on empirical relationships obtained from experimental studies or actual plant operation. This is generally true of the field of drying wherein a satisfactory theoretical development has not yet been accomplished. Several attempts have been made at various times to develop a theory of drying based on the flow of moisture within the material. The most widely known development of this kind is the diffusion theory of drying. An early development by Libman (5)* proposed a theory of porous flow for solid materials. His work, however, reduced essentially to the diffusion theory, since he assumed that the rate of capillary flow was proportional to the moisture concentration gradient which is identical with the diffusion theory. Other workers (1, 10) have presented a theory of capillary flow which holds generally for granular materials, and partially for other substances. Hougen, McCauley, and Marshall (3) have pointed out the limitations of the diffusion theory in drying, and they indicated those cases in which diffusion is likely to occur.

SUMMARY — The importance of food drying has predominated in times of war, present practice in the field having received its development and expansion during the Boer War and World War I. The great need for large supplies of concentrated foods for our armies and those of our allies — and our civilian allies — has created a tremendous demand for dried foods. . . . There is a big field for more engineering attention to food drying problems, and heating, air conditioning, and piping are all involved. For this reason, Heating, Piping & Air Conditioning presents in a group of articles —of which this is the third—a comprehensive and practical review of food drying today. A description of the general problem of food drying was given in Mr. Marshall's first article, published in the September HPAC. In October, he covered methods and equipment, and this month he gives data on the fundamentals of drying. . . . The author has specialized in the subject of drying, and is connected with the technical division of the engineering department at E. I. du Pont de Nemours & Co.

The conclusion to draw from all of these theoretical studies is that it is fallacious to assume that internal moisture movement within a material undergoing drying can be simply defined. In other words, the flow of water may be caused by a combination of many factors, some of which are capillarity, pressure due to shrinkage, true diffusion of liquid moisture, the force of gravity, and a sequence of vaporizations and condensations within the solid. Because of this complexity of internal moisture flow and the tremendous number of solid materials with different internal structures, no satisfactory theory has yet been developed for predicting the rate of this flow and hence the rate of drying. As a result, the fundamentals of a practical approach to drying have been based on the external drying conditions; i.e., the temperature, humidity, and velocity of the drying air, and the physical condition of the material being dried. Nevertheless, a drying problem should properly be analyzed from the standpoint of the nature of the solid itself as well as on the basis of the external conditions, since it is the nature of the flow within the material, i.e., the material characteristics, which determines what the drying conditions should be.

In the drying of foods, it would be expected that internal moisture movement would be caused by capillarity, shrinkage, and diffusion. The importance of shrinkage in food drying is indicated by the marked difference in volume between dried foods and wet foods. Capillarity would be expected to occur *between* cell walls, while diffusion would occur *through* the cell walls. This situation is similar to the flow of moisture in wood.

Design Fundamentals

The design of a dryer requires information on the following items:

A material balance.
A heat balance.
The rate of drying under the given conditions of air flow, temperature, and humidity.

The Material Balance—The material balance involves an accounting of the material streams and their components. Thus, the difference between the moisture content of the inlet and outlet food is balanced against the moisture picked up by the drying air. It is assumed that the rates of air and material flow are accurately known. The balance on the water may be written in words as follows: Water in entering air + water in entering food = water in air leaving + water in food leaving; or more conveniently, by symbols,

$$w_a H_1 + W_1 F = w_a H_2 + W_2 F \quad ..[1]$$

where w_a = lb of dry air per hr; H = lb water per lb dry air; F = lb of dry food per hr; and W = lb of water per lb dry food. Subscripts 1 and 2 refer to entrance and exit conditions, respectively.

Equation 1 may be written in the more useful form,

$$w_a (H_2 - H_1) = F(W_1 - W_2) \quad ..[2]$$

The use of the weight rate of air flow is convenient in drying calculations, since it eliminates the necessity of correcting for temperature variations and permits the use of values of absolute humidity.

The material balance, equation 2, is useful, primarily, in analyzing actual dryer performance, and it is not utilized to any great extent in design calculations, since the terms of the equation are generally determined by other means. It is of value, however, in analyzing the final design to determine the permissible amount of air recirculation, or to ascertain whether a proper design has been made from the standpoint of the saturation of the drying air.

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*Numbers in parentheses refer to references at end of the article.

The Heat Balance—The heat balance is extremely important in the design of a dryer. By means of it the required air quantities can be determined, as well as the size of the heating units. For drying calculations, the heat balance may be written as heat given up by air per hr = latent heat of evaporation per hr + sensible heat gained by stock per hr + sensible heat per hr to bring water to evaporation temperature + radiation losses per hr. Or, by symbols,

$$w_a c_s (t_1 - t_2) = \lambda_w F(W_1 - W_2) + cF(t_2' - t_1') + F(W_1 - W_2)(t_e' - t_1') + \text{radiation losses per hr.} \qquad [3]$$

where c_s = average humid heat of wet air, Btu per lb dry air per degree Fahrenheit = $0.24 + 0.45\,[(H_1 + H_2)/2]$; t = air temperature, degrees Fahrenheit; λ_w = latent heat of evaporation of water, Btu per lb; c = specific heat of dry solid; and t' = temperature of wet solid. Subscripts 1 and 2 refer to entering and leaving conditions respectively, and e refers to evaporation temperature.

For calculation purposes, it is often convenient to include the sensible heat term for water with the term for latent heat and to use a slightly higher average figure for λ_w of about 1100 Btu per lb water.

In equation 3, the following factors are generally known: Rate of feed, F. Initial and final moisture contents, W_1 and W_2. Specific heat of the solid, c, and the average humid heat, c_s. The permissible entering air temperature, t_1. The entering material temperature, t_1', and the allowable exit material temperature t'_2.

The quantities to be determined are w_a and t_2. In the design of some dryers, such as rotary or tunnel types, it is customary to determine t_2 first and then to calculate the air rate, w_a. In other types of dryers, such as tray dryers or through-circulation dryers in which a fixed air velocity is maintained, w_a is calculated first and then t_2. It is not generally possible to write a rigorous analytical expression for t_2 because of the many variables involved. This is due to the fact that t_2 depends upon economic factors as well as on the drying conditions. It is a function, in part, of the drying rate, the cost of heat, the amount of recirculation possible, and the economical size of the equipment. In many instances, t_2 is arrived at either on the basis of past experience or on the basis of experimental drying tests.

The Drying Rate—This item is the most important factor in a drying problem, because it determines the size of the dryer to be used. In contrast to the heat and material balances, which are equilibrium conditions, the rate of drying is a kinetic condition, and as such involves the time element.

In an industrial drying problem, an attempt is made to obtain the fastest possible drying rate which is compatible with the given conditions of temperature, air rate, amount of moisture to be removed, and the properties of the material to be dried.

In order to understand the significance of the rate of drying, it is well to review briefly the fundamental manner in which a solid material dries. If, for any given conditions of drying, a plot is made of the moisture content of the solid vs. time, a curve of the general nature shown here as the drying *time* curve is obtained. This curve may be considered as generic for the drying of all solid materials, and it may be readily determined in a suitable experimental dryer. If one were to differentiate the curve and plot the derivatives vs. the moisture content, a curve similar to the drying *rate* curve given here would be obtained; the ordinate in this case is the drying rate, which may be expressed as lb of water evaporated per hr per sq ft, or lb of water evaporated per hr per lb of dry stock. The latter expression results directly from the differentiation if moisture contents are expressed on the dry basis. If the moisture content is expressed on the wet weight basis, the resulting time curve has no linear portion.

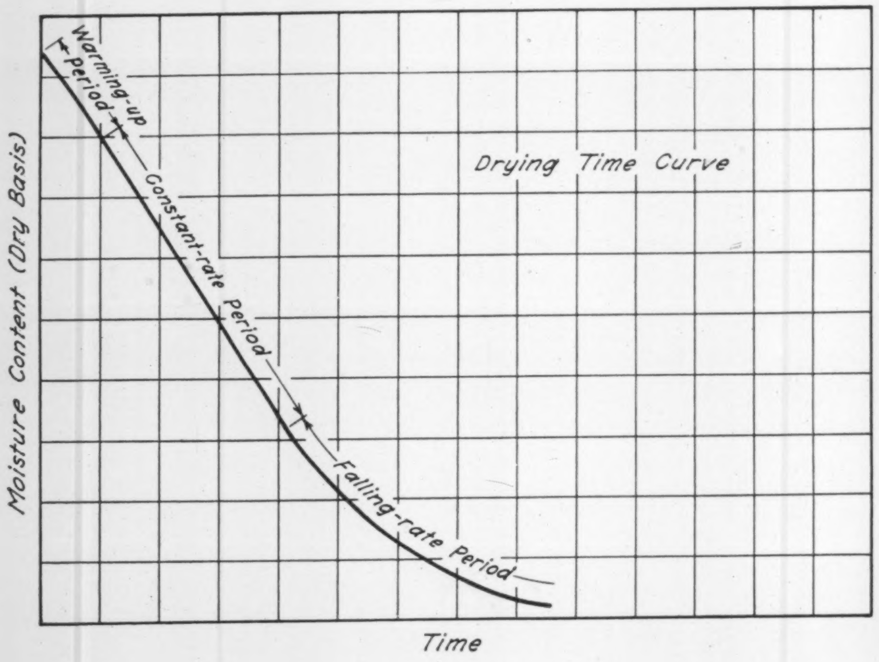
The following interpretation is generally given to the drying *rate* curve: The section BC is attributed to a warming-up period during which the surface of the solid is reaching a uniform temperature. This temperature is maintained constant during the period of a constant drying rate shown by AB, and indicated by the linear portion of the drying *time* curve. The drying rate in this period is determined by the air velocity, air temperature, air humidity, and the surface area exposed to evaporation. All these factors are external drying conditions.

It has been found that the constant rate on an area basis may be expressed by the following equation:

$$\frac{dW}{A\,d\theta} = k' G^n (p_s - p_a) \qquad [4]$$

where $dW/Ad\theta$ = lb of water evaporated per hr per sq ft; k' = mass transfer coefficient defined by equation 4; G = mass velocity of air, lb per hr per sq ft; p_s = vapor pressure of water vapor at the surface temperature of the solid; p_a = partial pressure of water vapor in the drying air; and n = constant.

At low surface temperatures (115 degrees Fahrenheit and under), an error of 10 per cent or less in the driving force is involved if, instead



of vapor pressures, the approximate relationship

$$H = \frac{18}{29}\,\frac{p}{P}$$

or $p = 1.61H$, for $P = 1$ atmosphere, is substituted in equation 4 to give

$$\frac{dW}{A\,d\theta} = k''\,G^{n}\triangle H \ldots\ldots\ldots\ldots [5]$$

where $k'' = 1.61\,k'$; $\triangle H = H_s - H_a$; H_a = humidity of the ambient air; and H_s = saturation humidity corresponding to the temperature of the drying surface.

The value of the exponent n has been found by various workers (4, 6, 8) to range from 0.6 to 0.8. Obviously, equation 4 expresses the rate of drying in terms of the external drying conditions, and it indicates the effect on the constant rate when any of these conditions are varied. H_s, the surface humidity, under adiabatic conditions, corresponds to the saturation humidity at the wet bulb temperature of the drying air. For calculation purposes, this is often assumed to hold when actual data are lacking. Radiation, however, often makes this assumption untrue. Shepherd, Hadlock, and Brewer (8) have presented a chart for predicting the constant drying rate for materials dried in trays.

Referring again to the drying *rate* curve shown here, *AD* represents the first stage of the falling-rate period, and when this portion of the rate curve is linear it is generally assumed that the surface, although still wet, is gradually drying out. The moisture content corresponding to point *A* is termed the *critical* moisture content. The portion of the rate curve corresponding to *DE* represents the period in drying where the outer surface has become dry, and subsurface evaporation is taking place.

For hygroscopic materials, drying cannot be carried below the equilibrium moisture content corresponding to the drying conditions. This point is represented by *E* on our curve.

A more detailed discussion of the drying rate curve has been given by Sherwood and Comings (9) and should be referred to as a supplement to the brief discussion given here.

In the falling-rate period, the drying rate is a function of the free moisture content and may be expressed as

$$\frac{dW}{A\,d\theta} = \alpha f(W) \ldots\ldots\ldots\ldots [6]$$

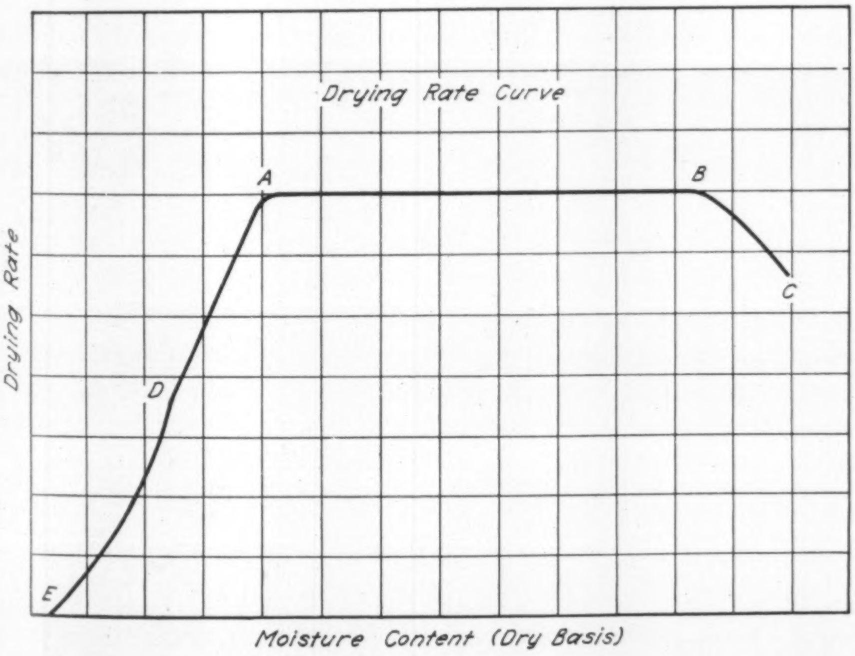
When the nature of the function *f(W)* is known, equation 6 can be integrated to determine the drying time in the falling-rate period. The simplest case is when $f(W) = aW$. Then, equation 6 can be integrated to give

$$ln\frac{W_c}{W} = \alpha' A(\theta - \theta_c)$$

$$\text{or } W = W_c e^{-\alpha' A(\theta - \theta_c)} \ldots [7]$$

where A = area of drying surface; W_c = critical moisture content; W = final moisture content at time θ; $\alpha' = a\alpha$; θ_c = time required to reach W_c (α = constant of proportionality and a = a constant; e = base of natural logarithms).

The usual test to determine whether or not equation 7 holds for a given material is to plot W vs. θ on semilogarithmic paper. If equation 7 applies, this plot will give a straight line from which a' may be determined.



VICTORY CENTER

"Ideas can help to win the war the American way, and it is vitally important to keep open the channels whereby ideas flow." That is the belief of a group of citizens who are sponsoring Victory Center, 745 Fifth Ave., New York City, a nongovernmental clearing house for civilian ideas designed to increase public safety, build morale, and improve the efficiency of the nation at war. The center has been created as a central place for receiving and testing new ideas and projects.

[Note: The foregoing discussion will be used in a future article by Mr. Marshall to analyze data on drying and illustrate design procedures.]

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Ventilating System Supplies 700,000 Cfm for Spray Painting Bomber Planes

One of the unusual features of two recently completed bomber assembly plants is the provision made at each plant for a paint spray structure large enough to permit the painting—after complete assembly—of huge bombing planes. The enclosure in which the assembled planes are painted is 150 ft wide, 200 ft long and 35 ft high. A partition which can be lowered from the ceiling to the floor makes it possible to divide the enclosure into two sections, each large enough to handle an entire plane.

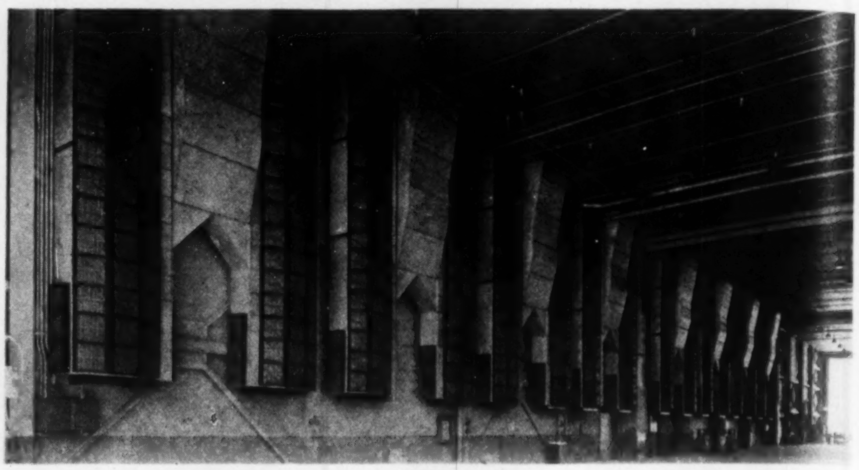
There is a ventilating system to provide a means of exhausting the paint laden air, and replace it with washed and filtered air. Such a system is required to protect those handling the spray nozzles from spray and fumes and to eliminate fire and explosion hazards.

The ventilating problem was solved by designing a system which draws 700,000 cfm of air from the outside atmosphere, washes it, warms it, filters it, draws it across the enclosure in an even flow, washes out the pigment, and exhausts the air to outside. There are eight air supply units and eight exhaust units.

For every pound of paint sprayed, there is supplied 5250 lb of air and 6466 lb of water. This is about five times the paint-air-water ratio of comparable commercial installations.

All equipment for the air supply units is mounted in the roof trusses, taking outdoor air from near the roof line. Fans draw the air in through automatic inlet louvers. After being tempered by steam coils, washed, and reheated to a constantly controlled temperature of 80 F, the air is evenly distributed over the entry wall through a system of dampered ductwork.

Admission of the air into the spray enclosure is then made through replaceable glass fiber air filters. These filters consist of mats of fine, interlaced glass fibers coated with a dust catching adhesive. Their function is to strain out any particles of dust or pollen that may remain in the air stream. They are arranged in 16 vertical bay panels, each containing 57 filter cells. Each cell provides room for four filters.



Glass fiber air filters arranged in 912 cells, each cell containing four filters, line the wall through which air enters the paint spray enclosure at each of two bomber assembly plants

The ductwork and filters are so arranged that the clean air enters the enclosure in such a way as to provide an evenly distributed flow from side to side, from floor to ceiling. The overspray from the paint spray guns is carried horizontally by the flow of air from the point where it is produced, across the width of the enclosure, to the nearest exhaust opening at the opposite side. The lighter overspray has no opportunity to settle in traversing the enclosure.

The eight exhaust units are arranged to draw the air out at three levels, thus helping to maintain even distribution. Each section of each unit has a constant flow of water over its flood sheet, the surface exposed to the room.

A suction draft exhaust fan, situated outside of each unit, draws the paint laden air through a series of conical, high pressure water sprays, confined in circular orifices. Centrifuge tubes completely separate the water and sludge from the air and deposit them in a sludge tank below. Only clean air passes through the exhaust ducts and fan.

The sludge tank for each unit holds 2000 gal of water. The water in these tanks is continuously recirculated through the units by pumps mounted on the floor in the spaces between them. The only loss of water is from evaporation. Water spray nozzles are in plain sight and readily accessible. There are no concealed eliminators that require removal for cleaning.

The glass fiber filters through which the clean air enters at the opposite side of the enclosure do not support combustion, and would tend to act as a barrier to fire spreading into the supply system. Further, the dust catching adhesive with which the glass fibers are coated carries underwriters' approval as a material without smoke or fire hazard.

The entire system is controlled by electric pushbuttons within reach from the paint floor level. Each of the supply and exhaust units is independently controlled. Any combination of the units can be operated together. Automatically controlled dampers on both the supply and exhaust units close when a unit is shut down and open when it is started up again. When no painting operations are in progress, but the outside temperature requires heat in the structure, about 10 per cent of the capacity of the supply air is drawn in through one unit to maintain a working temperature.

Glass fiber material is used throughout the walls and roofs of these plants as thermal insulation.

HEATING, PIPING & AIR CONDITIONING DATA SHEET

Steam Requirements of Process Equipment

Data on the steam requirements of process equipment are essential in the design and operation of steam generating equipment, piping systems, and in the work of district heating engineers. The information given here was collected from a number of sources and given in the most recent report of the commercial relations committee of the National District Heating Association, prepared by R. M. McQuitty, utilization engineer, Union Electric Co. of Missouri. The figures are presented as a compilation of available data, and not to establish a standard of steam requirements.

The columns headed "Authority" list the sources of the figures. District heating companies contributing data included The Detroit Edison Co., Detroit, Mich. (No. 1), Union Electric Co. of Missouri, St. Louis, Mo. (No. 2), Northwestern Electric Co., Portland, Ore. (No. 3), Consolidated Gas Electric Light and Power Co., Baltimore, Md. (No. 4), Board of Water and Electric Light Commissioners, Lansing Mich. (No. 5), and Philadelphia Electric Co., Philadelphia, Pa. (No. 6).

Publications from which data were taken and where further information may be found include the NDHA Handbook, 1921 ed., pp. 1131, 1132, 1267, and 1291 (A); Hoffman Specialty Co. Data Book, p. 168 (B); Warren Webster & Co. Bulletin 1200E, 1936, p. 20 (C); NDHA Proceedings, 1939, pp. 165-167 (D); NDHA Handbook, 1932 ed., p. 431 (E); and NDHA Proceedings, 1940, p. 162 (F).

STEAM REQUIREMENTS OF LAUNDRY AND KITCHEN EQUIPMENT

	Operating Pressure, Psi Ga	Lb per Hr Steam Use, Max	Lb per Hr Steam, in Use	Lb Steam per Week	Lb Steam per Month	Authority
Cleaning and Pressing Equipment						
Clothes Presses						
U. S. Hoff-Man (electric vacuum)	18	...	25	1,000	....	1
Excelsior Mach. Corp. Mod. 460 (electric vacuum)	30	...	...	1,700	....	1
U. S. Hoff-Man (electric vacuum)	30	...	25	800	....	1
U. S. Hoff-Man Mod. U.C.O.-5 (steam operated vacuum)	50	67.2	38.1	....	....	2
Steam vac. jet on Hoff-Man press. Venturi type, max instantaneous use (jet only)	50	58.0	...	....	....	2
Large press. (steam vacuum)	65	...	30	....	....	D-166
Large press. (electric vacuum)	65	...	20	....	....	D-166
Equipment in Hotel Laundry						
1 dry tumbler, 54"x70", Armstrong	95	70	...	....	....	3
1 mangle, 6 roll, American, 110"	95	380	...	....	....	....
Equipment in Hand Laundry						
Standard press, Prosperity, (Use based on avg. of five presses). Unit use per press	95	...	10.13	....	....	....
Bosom Press, Pantex, Mod. L-410-P-LHD7	95	...	32.6	....	....	....
Collar & cuff press, Pantex, Mod. L-453-P-LHD7	95	...	34.7	....	....	2
Sleever Huebsch No. 5857	95	...	11.2	....	....	....
Total steam use of all equipment = 3.71 lb steam per lb. laundry						
Miscellaneous Laundry Equipment						
Tumbler (drying machine)						
Size 40"x94"	80-100	...	360	....	....	D-167
Size 30"x36"	80-100	...	225	....	....	D-167
Mangle (flat work ironer)						
Size 8 roll	80-100	...	480	....	....	D-167
Size 6 roll	80-100	...	415	....	....	D-167
Large shirt body press	80-100	...	190	....	....	D-167
Cuff and neck band press	80-100	...	15	....	....	D-167
Standard press	80-100	...	105	....	....	D-167
Feather cleaning and sterilizing (Cap. 15 pillows per hr)	80-100	...	175	....	....	D-167
Group Use of Landry in Large Hotel (400 *rooms*)						
1 mangle, Amer. Laundry						
5 presses						
1 sleever						
1 sock drier						
1 dry tumbler						
Total group use	90	...	...	8,350	334,000	5
Line loss only	90	...	...	....	12,300	5
(For further general data on laundry equipment, see A-1267.)						
Kitchen Equipment						
Steam jacket kettles						
40 gal cap. American Aluminum Co.	8	55	...	1,575	....	1
Full jacket cast iron kettle at various steam pressures	2	62	24	....	....	A-1131
	5	133	36			
	10	174	97			
	15	228	116			
Half jacket, cast iron kettle at various steam pressures	2	38	16	....	....	A-1131
	15	43	24			
Candy kettles						
Per sq ft of jacket	30	60	...	....	....	D-165
Per sq ft of jacket	75	100	...			
Dish warmers						
Size 5'6"x9"x5'4", Htg. Surface, 25.02 sq. ft. Htg. Element, Inside Temp. 149F	2	16	15	....	....	A-1131
Dishwarmer	..	...	30	....	....	E-431
Dishwarmer	..	...	20	....	....	E-431
Steam Tables						
Size 3'3"x1'9"x5" deep, htg. surface 0.99 sq ft	2	31	14	....	....	A-1131
Same table at 5 psi Press	5	32	20.5			
Size 9'0"x22"x8", 3.33 sq ft copper element	8	70	...	....	....	1
Size 6'9"x28"	..	...	24	....	....	E-431
Size 9'0"x28"	..	...	32			
Compartment Cookers						
Two compartment, leaking valves	2	...	...	8,890	....	1
Three compartment cooker	2	...	...	6,300	....	1
One compartment cooker.	2	...	...	4,200	....	1
Three compartment, new type	2	...	...	1,498	....	1
Three compartment trapped with closed vents	5	111	...	575	2,300	2
Three compartment	..	...	75	....	....	B-168
Compartment cooker	..	...	100	....	15,000	D-165
Bain Marie						
Size 36"x—	15-20	...	50	....	....	D-165
Size 9'0"x18"	..	...	41	....	....	E-431
Ice Cuber	30	80	...	1,945	....	1
Coffee and Hot Water Urns						
Test on 8 gal urn, 1.66 sq ft. brass element at various pressures	5	46	10	....	....	A-1132
	10	85	40			
	15	120	75			
3.32 sq ft brass element at various pressures	2	43	13	....	....	A-1132
	5	75	25			
	10	144	57			
	15	150	75			
	25	200	139			
Dishwashers						
Crescent, manual steam control	30	...	...	1,400	....	1
Hobart Mod. AM-5	10	...	...	840	....	1
—Dishwashing machine	15-20	...	70	....	1,500	D-165
—Dishwashing machine	..	...	60	....	....	E-431
Corned Beef Cooking Vat						
Vat size 10'0"x4'6"x3'0" 9 sq ft stainless steel element. Tested while cooking 1500 lb of corned beef	35	...	86	....	....	2
Group Equipment Use of Restaurant						
1 large steam kettle						5
1 four compartment vegetable cooker						
9 coffee urns						
3 steam tables						
Group steam use	45	...	...	37,500	150,000	
Line loss	45	...	...	....	13,200	
For further general data on restaurants, see						F-162 A-1267

STEAM REQUIREMENTS OF INDUSTRIAL PROCESSES

	Operating Pressure, Psi Ga	Lb per Hr Steam, Max	Lb per Hr Steam, in Use	Lb Steam per Week	Lb Steam per Month	Authority
Miscellaneous Factory Processes						
Plastic Pressing Bakelite Bottle Tops						
1 large and 2 small machines operating 24 hr per day. Entire uses (all equipment)	150	.	..	....	100,000	D-167
Vulcanizing (rubber stamps)						
1 press 12" x 12" operating 8 hr per day, total steam use	80	..	..	....	15,000	D-167
Steam Consumption in Tannery						
Lime vat (cap. 800 gal per vat) per set of 3 vats...	..	697	..	....	.....	A-1291
Paddle vat (cap. 1000 gal)	..	448	..	....	.....	
Revolving drums	..	531	..	....	.....	
Iron plate on press. 9" coil plate is 2'6" x 2'6" at 150 F	..	100	..	....	.....	
For further general data on above tannery, see						A-1291
Clothing Mfg. Equipment						
Mfg. of Women's Dresses						
22 steam-electric irons with standard cast iron separator on each iron	35	..	...	....	.....	2
2 sleevers, steam-electric type, with blower	35	.	...	....	.....	
4 presses, Pantex Mod. 41-0-C15	42	..	...	....	.....	
No open end steam.						
Piping, 205 ft (lineal) of 1", and 90 ft of 1½".						
127 ft of ¾"—all insulated.						
Total steam use*	..	291	264	....	65,100	
Individual use per steam-electric iron*	35	..	8.4	....	.....	
Individual use per press*	42	..	12.2	....	.....	
Individual use per sleever*	35	..	5	....	.....	
Total steam use of customer (average per mo)					65,100	
Mfg. of Men's Clothing						
16 presses (with central elec vac)						4
3 shoulder presses						
2 collar presses						
4 edge presses						
5 body presses						
2 back presses						
16 operated by five men**	65	.	122	4,880	.....	
8 under presses, central-electric vacuum**	65	..	103	4,120	.....	
120 lineal ft, 2", insulated main**	65	..	37	1,480	.....	
Avg unit use per press	65	.	7.6	....	.....	
Avg unit use per under press	65	..	12.9	....	.....	
Total steam use of customer (average per mo)					52,200	
13 presses (central elec vac)						4
2 shoulder presses						
2 collar presses						
2 edge presses						
5 body presses						
2 back presses						
13 operated by four men**	65	..	175	7,000	.....	
5 under presses (central elec vac)**	65	..	91	3,640	.....	
90 lineal ft. of 1½" insulated main**	65	..	25	1,000	.....	
Avg unit use per press**	65	..	10.9	....	.....	
Avg unit use per under press**	65	..	18.2	....	.....	
Total steam use of customer (average per mo)					99,300	
Cloth Sponging						
3 sponging machines						
9 refinishing cylinders						
1 manifold						
1 drier						
2 wetting-out tanks for cold water shrinking						
2 semi-decating units for high finish work						
Total load (8 hr per day)	50	..	1,800	....	225,000	D-165
1 sponging machine						
1 refinishing roll						
1 semi-decating unit						
1 manifold						
1 wetting-out tank for cold water shrinking						
1 drier						
Total load	90-100	..	1,000	....	100,000	D-165
1 under press	65	..	20	....	.....	D-166
1 steam iron (standard)	..	..	15	....	.....	D-166
Hosiery Dyeing						
24 hosiery drying forms—total per table of 24 forms	40	..	60	....	.....	D-166
8 twenty-four form, hose drying tables						
4 300 gal drying tanks						
Total load of business (8 hr per day)	40	..	....	....	60,000	D-166

---

*60 hours use per week.

**43 hours use per week.

APPROXIMATE UNIT STEAM CONSUMPTION OF KITCHEN APPLIANCES

	Lb per Hr of Steam
Stock and vegetable kettles (per 5 gal)	20.0
Coffee urns (per gal)	3.4
Water urn (per gal)	5.0
Steam tables (per sq ft)	1.7
Plate and cup warmers (per 20 cu ft)	35.0
Vegetable steamers (per compt.)	40.0
Soup warmer, 30"x30"x28"	100.0
Clam, lobster, and potato steamers (per compt.)	40.0
Egg boilers (3 compts.)	18.0
Oyster pots	18.0
Bain Marie (per sq ft)	3.4
Food Warming ovens (per 20 cu ft)	35.0
Silver burnisher and washer	69.0
Dishwasher (per tray)	60.0

Note: Above figures (from Reference No. 6) represent average operating conditions after warm-up period and do not include hot water.

Pressures will vary from 7 to 20 psi ga.

STEAM REQUIREMENTS OF HOSPITAL EQUIPMENT

	Cap. Bottles / Size / Cap., gal.	Depth of Water	Operating Pressure Psi Ga	Lb per Hr Steam, Max	Lb per Hr Steam, in Use	Authority
Hospital Equipment						
Sterilizers (non-pressure type)	Cap. Bottles	Depth of Water				
For bottles or pasteurization start with water at 70F, then maintained at boiling for period of 20 min.	36	3"	40	36	36	C-20
	54	3"	40	51	51	
	72	3"	40	69	69	
For instruments and utensils	Size					
Start with water at 70, then boil vigorously for period of 20 min.	8" x 9" x 18"	3½"	40	27	27	C-20
	9" x 20" x 10"	3½"	40	30	30	
	10" x 12" x 22"	4"	40	39	39	
	12" x 16" x 24"	4"	40	60	60	
	10" x 12" x 36"	4"	40	66	66	
	16" x 15" x 20"	10"	40	92	92	
	20" x 20" x 24"	10"	40	144	144	
Sterilizers (pressure type)	Size					
For surgical supplies. Sterilizing period, 30 min at temperature of 240-250 F	12" x 20"		40	22	22	C-20
	14" x 22"		40	28	28	
	16" x 24"		40	38	38	
	16" x 36"		40	54	54	
	20" x 28"		40	60	60	
	20" x 36"		40	78	78	
	20" x 48"		40	98	98	
	20" x 60"		40	124	124	
	10" x 20"		35-60	9.5	9.5	D-166
	24" x 48"		35-60	50.0	50.0	D-166
For instruments, sterilizing period, 20 min. at 240-250 F	12" x 20"		40	48	48	C-20
	14" x 22"		40	60	60	
	16" x 24"		40	72	72	
Sterilizers (pressure type) (autoclave)						
Sterilizing period, 30 min. at 240-250 F	15½" x 24"		40	24	24	C-20
	17½" x 26"		40	32	32	
	21½" x 30"		40	40	40	
	24" x 36"		35-60	42	42	D-166
Water Sterilizers	Cap., gal.					
Start with water at 70 F and maintain at temperature of 240-250 F for 15 min.	6		40	76	76	C-20
	10		40	120	120	
	15		40	180	180	
	25		40	300	300	
	50		40	600	600	
Unit requirements for water sterilizer = 2.5 lb steam per gal sterilized			35-60	..	..	D-166
Mattress Disinfector						
Size 30" x 42" x 84"			35-60	..	42	D-166
Size 60" x 66" x 108"			35-60	..	318	D-166
Blanket Warmer						
Size 18" x 24" x 72"			35-60	..	4	D-166

Highlights of Detroit Edison's Air Conditioning Health Survey

Henry S. Walker Summarizes Results of a Study Involving 1500 Employees

THE COMFORT advantages of air conditioned quarters have long been recognized. But what are the effects of air conditioning on the frequency of common colds?

For a number of years there have been controversial opinions among people: Some have contended that air conditioning is harmful to health, while others have stated that it is beneficial. Our study was devised to provide data that would help us to answer this question impartially and intelligently.

Fifteen hundred Detroit Edison employees cooperated with the research department in measuring this health aspect of air conditioning. Their answers to the question proved conclusively that air conditioning as practiced in the service building and general offices has no harmful effects on the health of company employees, as measured by the occurrence or severity of colds.

SUMMARY—Fifteen hundred employees cooperated in a study made of the health benefits of air conditioning by the research department of The Detroit Edison Co. Mr. Walker, supervising engineer of the research department, summarizes the highlights of the survey here, in an article taken from the Synchroscope, The Detroit Edison Company's employee publication. . . . Employees in both air conditioned and non-air conditioned quarters were involved in the survey, and certain pertinent data from sources other than employees were also obtained. The air conditioning of the service building has been described in articles in the October, 1937, and April, 1938, issues of HEATING, PIPING & AIR CONDITIONING; the system now serving the general offices building was described in HPAC for February, 1942

The study was started early in 1938, about five months before the air conditioned service building was completed. It ran spring, summer, fall, and winter for approximately 140 consecutive weeks — almost three years. At the start, about 1500 employees from various locations volunteered to participate in the study. Some employees were to occupy the new quarters, the rest were to remain in the then non-air conditioned general offices, where older heating and ventilating methods were in use. Since the study was completed, the general offices building has been completely air conditioned.

The first five month period prior to occupation of the service building was used as a control period to determine whether the two groups to be compared were alike as far as susceptibility to colds and coughs was concerned. If there had been a marked difference between the groups, the final results could not have been attributed to the one factor—air conditioning. Analysis indicated that the two groups were not significantly different with respect to respiratory infections, and as a consequence the comparisons could justifiably be made. During the survey, nearly 200,000 individual weekly reports were submitted by employees.

The participants, were, of course, subjected to conditions of the survey only 40 hr out of a 168 hr week. The exposure to infection during the non-working hours was not altered. Therefore it could hardly be expected that the survey would reveal any striking difference in the number or seriousness of infections in the two groups, unless the conditions in one building had been far better than in the other.

A great variety and volume of information was obtained with respect to the incidence of respiratory infections. It is too voluminous to report herein. The study included a comparative analysis of the following factors: Number of infections or colds, severity of these infections, time lost from work because of them, seasonal differences and epidemics, differences in the incidence of infections as between men and women, influence of age on occurrence of infections, and other factors.

All differences revealed between the two groups as a result of the study were tested by means of highly reliable mathematical procedures to assure that the differences were real and had not occurred on the basis of chance alone.

The most significant finding of the study on the basis of the survey was that each 100 employees in air conditioned quarters lost fewer days from work over the period of the study because of respiratory infections than did employees located in non-air conditioned quarters. However, this significant difference between the two groups was too small to warrant an unqualified statement in favor of air conditioned quarters. It was considered as evidence but not absolute proof.

Information separate from the data on respiratory infections was collected. Employees in air conditioned quarters reported more freedom from drafts than did employees in non-air conditioned quarters. Hay fever sufferers in the service building reported relief from their symptoms during working hours, while those in the non-air conditioned general offices building reported no such relief.

Certain pertinent data other than those submitted by employees also were obtained:

Dust: Rate of dust accumulation at representative locations in the service building was approximately one-quarter of that found at representative locations in the general offices building.

Bacteria: There was no significant difference in the bacteria counts in the two buildings.

Temperature: Temperatures maintained in the service building were decidedly more uniform than in the non-air conditioned general offices building.

Humidity: Humidity in the winter in the service building was maintained at a higher value than that prevailing in the non-air conditioned general offices building; in summer it was maintained at a lower value.

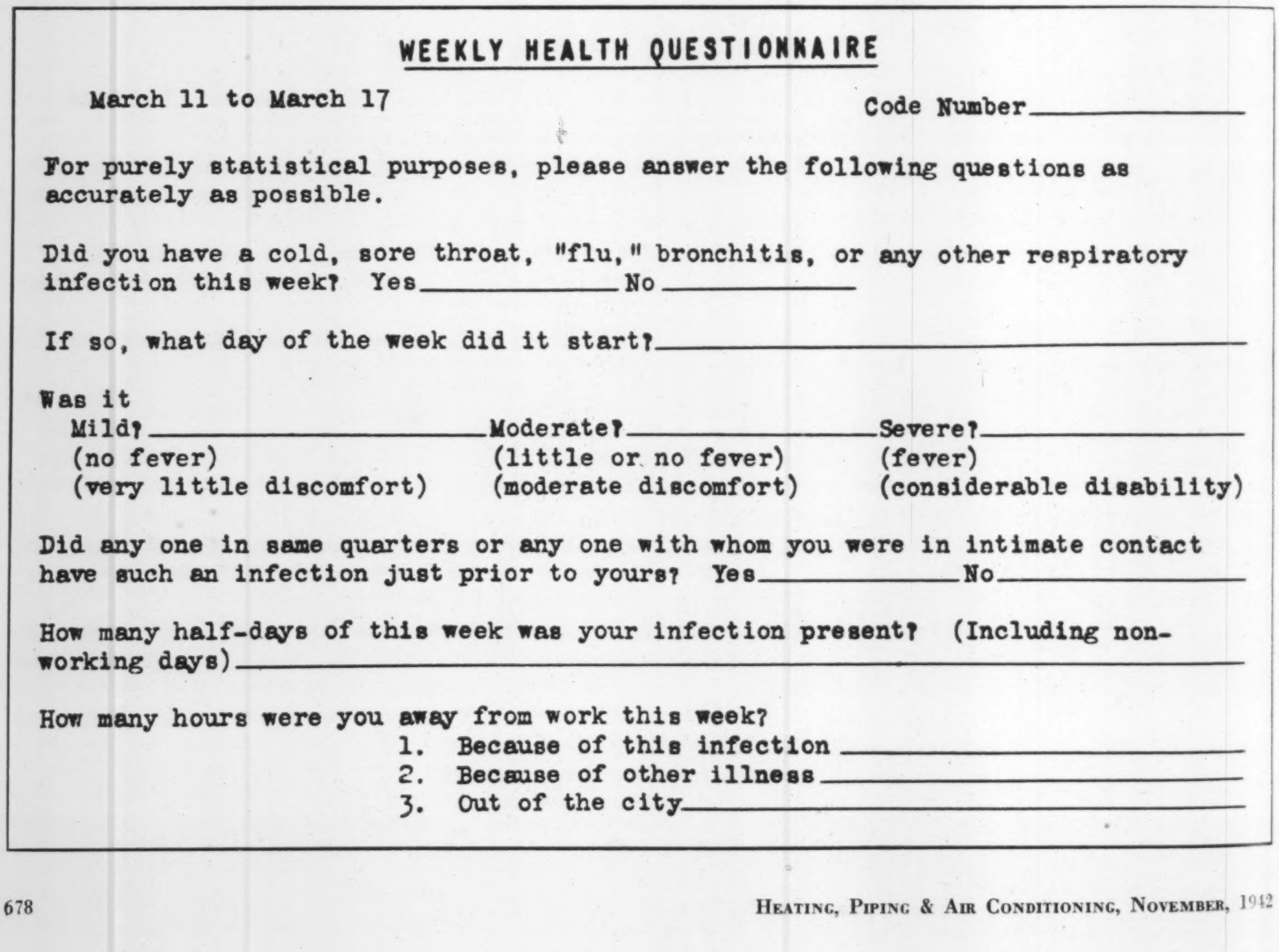
Pollen: Ragweed pollen counts in the service building were appreciably lower, seldom reaching a value that would be considered annoying to even a supersensitive person.

Since completion of this health survey, the general offices have been entirely air conditioned; therefore, the benefits in health and comfort in the air conditioned general offices now should be the same as in the air conditioned service building.

One of the most important facts brought out by this study was that beneficial results are obtained from year around air conditioning. The system in use in the service building (and the system now in use in general offices) is not just summer cooling. It is controlled heating or cooling, humidification or dehumidification as required, plus air filtering or cleaning at all times. The value in using air conditioning throughout the year was illustrated in the reports of increased comfort during summer months and improved health in the winter season.

Above—The air conditioned service building; note the use of glass blocks. . . . .

Below—The weekly health questionnaire form used in the study



WEEKLY HEALTH QUESTIONNAIRE

March 11 to March 17

Code Number____________

For purely statistical purposes, please answer the following questions as accurately as possible.

Did you have a cold, sore throat, "flu," bronchitis, or any other respiratory infection this week? Yes__________ No__________

If so, what day of the week did it start?____________________

Was it

Mild?__________	Moderate?__________	Severe?__________
(no fever)	(little or no fever)	(fever)
(very little discomfort)	(moderate discomfort)	(considerable disability)

Did any one in same quarters or any one with whom you were in intimate contact have such an infection just prior to yours? Yes__________ No__________

How many half-days of this week was your infection present? (Including non-working days)____________________

How many hours were you away from work this week?

1. Because of this infection ____________
2. Because of other illness____________
3. Out of the city____________

Heating·Piping·Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

MORE NAVY RESEARCH FOR SOCIETY

At the 48th Annual Meeting of the Society held in Philadelphia in January 1942, the Council passed a resolution offering without expense to the people of the United States through its government the full use of the ASHVE Research facilities, including the specially trained personnel of its Laboratory. Subsequently this resolution was transmitted to a number of government agencies and several of them expressed an interest in the proposal. On August 12 the United States Navy Department renewed the agreement with the Society to investigate several problems concerned with the basic design of heating, ventilating and air conditioning systems for naval vessels.

In renewing the agreement, the Navy Department stated that the results of investigations carried on by the Society under previous agreements had provided sound data which have had a marked effect in maintaining the fighting qualities of the officers and crew of naval vessels through improved design of heating, ventilating and air conditioning systems for vessels currently in operation.

The Navy further expressed the hope that the research work currently required will be carried out as expeditiously as is consistently possible, in order that several answers required may likewise be reflected in the design at the earliest possible moment.

The program assigned to the ASHVE Research Laboratory is now being actively pursued and personnel has been added to the staff to complete the work as soon as possible. Clark M. Humphreys, assistant professor of mechanical engineering at Carnegie Institute of Technology and former ASHVE Council member, was recently appointed acting director of the Laboratory to supervise and administer the project.

Another phase of the investigation is being conducted at the cooperating institution, Case School of Applied Science, under the direction of Prof. G. L. Tuve, a former ASHVE Council member and currently a member of the Guide Publication Committee of the Society.

F. C. McIntosh, Chairman
Committee on Research

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Printed in U. S. A.

Friction Heads Due to Water Flow in Copper, Brass and Other Smooth Pipes

By F. E. Giesecke,* College Station, Tex.

Summary—This paper presents a comparison of the experimental determination of friction heads in brass pipes, made in 1892 by John R. Freeman and published in 1941 by the *ASME*, with other experimental determinations, and the development of a formula for friction heads of 140 F water flowing in Type M copper tubes.

About 1902 a thorough and extensive study[1] was made at Cornell University of the resistances to the flow of water in pipes and the results of this study were presented at a meeting of the *American Society of Civil Engineers* on September 2, 1903. Among the pipes used by Saph and Schoder in their study were 15 brass pipes ranging in size from 2.090 to 0.0107 in.

In 1913 a German mathematician, published his classic study[2] of the friction in fluids and among the experimental data studied by Blasius were those of Saph and Schoder, which he pronounced the most carefully executed and the most extensive available. Of the 15 brass pipes tested by Saph and Schoder, Blasius used 8, ranging in size from 2.090 to 0.0107 in. For each test result he calculated Reynolds' number (R) and the value of f in the formula

$$h = f\frac{l}{d}\cdot\frac{v^2}{2g} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

and plotted f against R to logarithmic coordinates. From the resulting chart it appeared that a definite relation existed between f and R, and Blasius concluded that this relationship could be expressed, with sufficient accuracy, by the formula

$$f = 0.3164\,R^{-0.25} \quad \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

Of the experimental data secured by Saph and Schoder 36 were for brass pipes and used by Blasius in his studies as shown in Table 1 and in Fig. 1. The chart of Fig. 1 is drawn to semi-logarithmic coordinates in order that the values of f

Table 1—Tests by A. V. Saph and E. H. Schoder—1903

Experiment No.	Friction Head Ft per 100 ft	Temperature F	Velocity fps	f	R
Brass Pipe No. 2; Diam. = 0.1742 ft = 2.09 in.					
1	4.155	36.0	4.591	0.0222	44,700
11	4.377	49.5	4.893	0.0205	53,600
3	1.342	36.0	2.412	0.0260	23,600
29	0.892	69.8	2.063	0.0235	33,900
33	0.148	69.8	0.735	0.0308	12,070
716	0.812	38.5	1.825	0.0273	18,620
Brass Pipe No. 3; Diam. = 0.12484 ft = 1.498 in.					
194	17.72	68.8	8.994	0.01761	104,500
550	19.36	35.5	8.738	0.0204	60,300
39	3.44	69.1	3.568	0.0217	41,400
547	6.49	36.0	4.708	0.0235	32,800
203	0.291	71.8	0.870	0.0309	10,520
549	1.12	37.6	1.713	0.0306	12,310
Brass Pipe No. 4; Diam. = 0.10311 ft = 1.237 in.					
303	15.43	62.0	7.135	0.02015	62,100
721	14.53	37.2	6.516	0.02275	38,400
344	5.45	54.6	3.89	0.0239	30,500
723	4.99	38.0	3.542	0.0264	20,800
45	1.39	69.0	1.813	0.0281	17,450
724	2.12	38.5	2.167	0.0299	13,050
Brass Pipe No. 7; Diam. = 0.05251 ft = 0.630 in.					
196	61.10	69.8	9.899	0.0211	48,900
476	67.00	37.8	9.711	0.0240	29,500
53	12.61	70.0	4.028	0.0263	20,000
473	20.47	46.0	4.949	0.0283	17,300
388	0.345	85.8	0.538	0.0403	3,260
56	0.652	70.7	0.728	0.0416	3,650
Brass Pipe No. 9; Diam. = 0.03137 ft = 0.376 in.					
176	172.4	67.5	12.28	0.0231	35,000
514	170.1	37.7	11.415	0.0263	20,600
518	55.8	37.8	6.015	0.0312	10,900
179	40.9	69.3	5.334	0.0290	15,620
519	27.4	37.9	4.001	0.0345	7,270
284	14.08	67.0	2.903	0.0337	8,250
Brass Pipe No. 11; Diam. = 0.02347 ft = 0.282 in.					
274	208.6	57.5	10.872	0.0268	20,200
487	199.02	39.1	10.170	0.0291	14,170
483	58.35	37.1	4.975	0.0356	6,670
80	43.99	69.0	4.567	0.0318	10,000
139	8.89	70.0	1.832	0.0401	4,070
80	29.37	69.0	3.626	0.0348	7,950

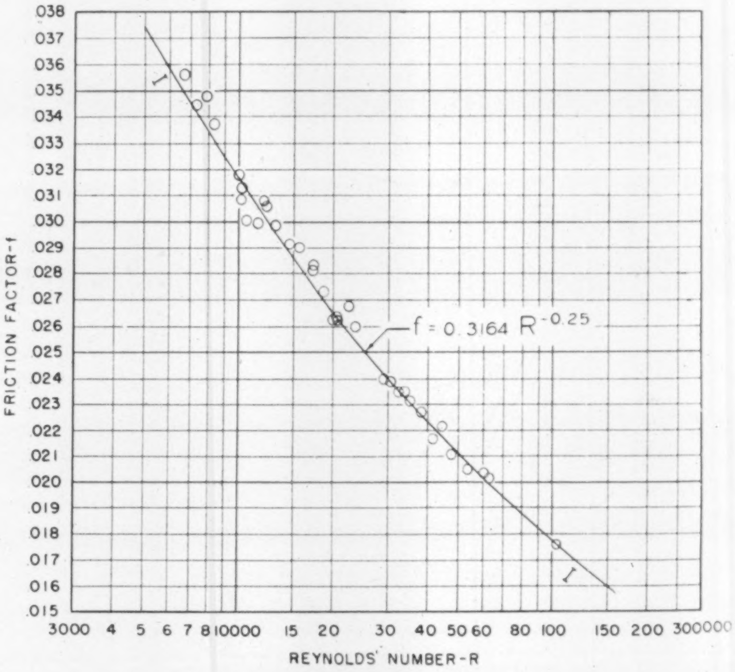

Fig. 1—The Saph-Schoder-Blasius f-R line

Thirty-six of the Saph and Schoder test results and Blasius' f-R relation: $f = 0.3164\,R^{-0.25}$ which is probably not accurate for values of R above 75,000.

---

*Professor Emeritus, Agricultural and Mechanical College of Texas. MEMBER of ASHVE.

[1] Results of investigations made by A. V. Saph and E. H. Schoder at Cornell University on the resistances to the flow of water in pipes. (Published in *ASCE Transactions*, Vol. LI, pp. 253-309.)

[2] Friction in Fluids, by Dr. H. Blasius, Hamburg. (Published in *Mitteilungen uber Forschungsarbeiten*, Vol. 131, pp. 1-40.)

For presentation at the 49th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cincinnati, Ohio, January 1943.

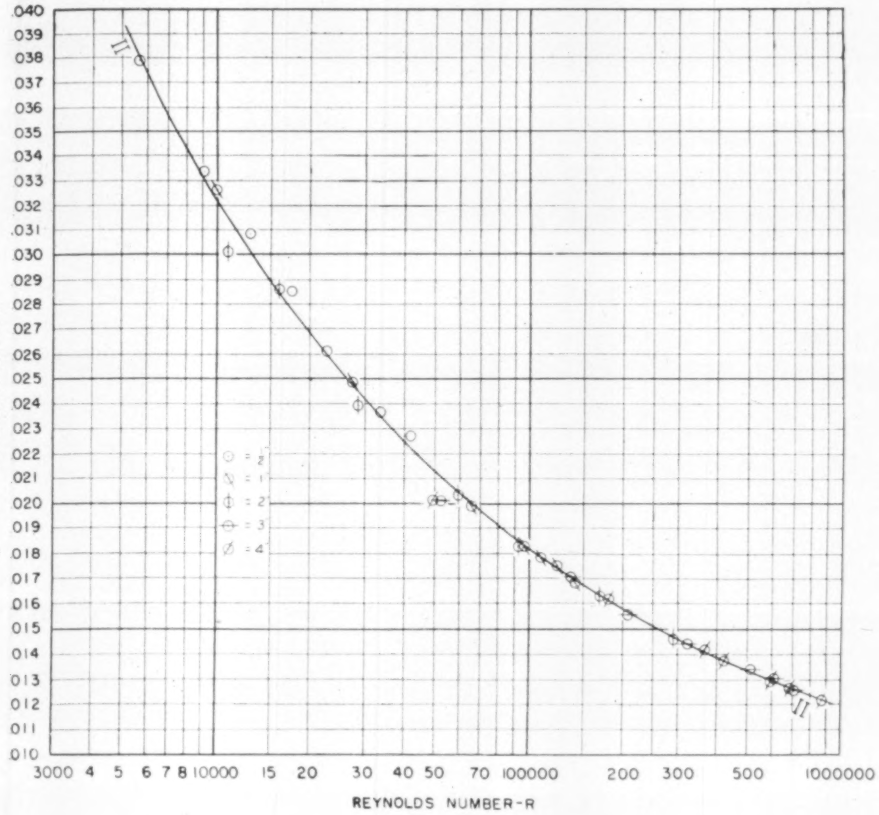

Fig. 2—The J. R. Freeman *f-R* line

Thirty-five of Freeman's test results and the line representing the *f-R* relation as determined by Freeman.

may be read easily and accurately. The line representing Blasius' formula, Equation (2) appears as a curved line on this chart instead of as a straight line as it would if the chart had been drawn to logarithmic coordinates.

A complete record,[3] with discussions, of the very valuable hydraulic research conducted by John R. Freeman in the Jackson Co. cotton factory at Nashua, N. H., and completed in 1892, 50 years ago, was published in 1941.

Among the pipes tested by Freeman were 5 brass pipes, ranging in size from 4 in. to ½ in. and varying in length up to 122 ft. These pipes were made up of pieces 12 to 15 ft in length and carefully joined at the ends with special sleeve couplings made of heavy tinned plate. Mr. Freeman states, *Rarely, if ever, in practical work could straighter lines of pipe be secured.* Of the experimental data secured by Freeman 35 are shown in Table 2 and in Fig. 2. Since the data shown on Fig. 2 range almost up to $R = 1{,}000{,}000$, while those of Fig. 1 do not range beyond $R = 100{,}000$, the character of the relationship between f and R can be determined more accurately from Fig. 2 than from Fig. 1. If Fig. 2 had been drawn to logarithmic coordinate, it would show clearly that the f-R relationship is not rectilinear and that Blasius' formula can be used only for a limited range of R values. In discussing this feature Clarke Freeman states:

> It is our opinion that, had my father's experiments been published even as early as 1900, it is doubtful Blasius would have ever published his straight-line friction-factor equation which has been so widely used and which is the basis of the *one-seventh power law* velocity-distribution equation, valid only over the range $R = 10{,}000$ to $R = 100{,}000$, according to the experiments of J. Nikuradse.

Studies described in a paper[4] published in 1930 relate primarily to

Table 2—Tests by J. R. Freeman—1892 and earlier

BRASS PIPE ½ IN.; AVERAGE INTERNAL DIAMETER 0.5470

EXPERIMENT No.	TEMPERATURE F	VELOCITY FPS	FRICTION HEAD FT	FRICTION HEAD FT PER 100 FT	f	R
498	68	10.22	49.65	81.44	0.02278	42,710
500	68.5	8.013	63.55	52.09	0.02370	33,680
502	68.5	5.448	32.37	26.53	0.02612	22,900
503	69	4.137	20.39	16.71	0.02853	17,510
504	69	3.064	12.09	9.913	0.03084	12,970
507	70	2.123	6.359	5.212	0.03377	9,107
509	71	1.305	2.695	2.209	0.03792	5,670

BRASS PIPE 1 IN.; AVERAGE INTERNAL DIAMETER 1.061

EXPERIMENT No.	TEMPERATURE F	VELOCITY FPS	FRICTION HEAD FT	FRICTION HEAD FT PER 100 FT	f	R
541	67.5	17.66	56.44	92.05	0.01686	143,100
542	67.5	15.42	44.81	73.09	0.01756	124,900
543	67.5	13.69	72.03	58.75	0.01789	111,000
544	68	11.79	27.39	44.69	0.01836	96,180
547	68	5.101	12.38	10.09	0.2215	41,620
549	68.5	3.359	6.031	4.919	0.02489	27,600
550	68.5	1.220	0.5219	0.8513	0.03264	10,000

BRASS PIPE 2 IN.; AVERAGE INTERNAL DIAMETER 2.107

EXPERIMENT No.	TEMPERATURE F	VELOCITY FPS	FRICTION HEAD FT	FRICTION HEAD FT PER 100 FT	f	R
529	66.5	3.760	1.558	2.550	0.02039	59,510
521	69	17.88	50.64	41.51	0.01468	292,600
524	69.5	10.31	18.71	15.34	0.01631	169,800
528	66.5	5.922	6.946	5.694	0.01835	93,670
531	66.5	1.814	0.8499	0.6967	0.02392	28,700
532	67	1.004	0.3115	0.2553	0.02863	15,980
534	67	0.6892	0.1596	0.1308	0.03111	10,980

BRASS PIPE 3 IN.; AVERAGE INTERNAL DIAMETER 3.083

EXPERIMENT No.	TEMPERATURE F	VELOCITY FPS	FRICTION HEAD FT	FRICTION HEAD FT PER 100 FT	f	R
599	66	30.88	44.84	73.00	0.01258	705,700
603	66	14.08	21.41	17.48	0.01449	322,000
607	66.5	26.31	67.26	54.92	0.01305	605,600
608	66.5	22.02	48.51	39.61	0.01343	506,900
615	67	2.273	0.8122	0.6632	0.02111	52,670
611	66.5	9.092	4.862	7.917	0.01574	209,200
613	67	5.986	5.563	3.725	0.01709	138,700

BRASS PIPE 4 IN.; AVERAGE INTERNAL DIAMETER 3.999

EXPERIMENT No.	TEMPERATURE F	VELOCITY FPS	FRICTION HEAD FT	FRICTION HEAD FT PER 100 FT	f	R
571	67.5	28.71	57.14	46.78	0.01217	873,700
574	67.5	19.65—	14.31	23.30	0.01294	597,900
577	68	13.74	14.79	12.11	0.01375	421,100
580	69	22.11	35.32	28.91	0.01268	686,700
583	69.5	11.80	5.603	9.223	0.01420	368,900
586	69.5	5.817	3.122	2.555	0.01619	181,900
588	70	1.583	0.3044	0.2492	0.02133	49,820

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[3] Flow of Water in Pipes and Pipe Fittings, by John Ripley Freeman, C. E., (edited by Clarke Freeman, son of J. R. Freeman).

[4] Hydraulic Service Characteristics of Small Metallic Pipes, by G. M. Fair, M. C. Whipple, and C. Y. Hsiao, Research Fellow, read before *New England Water Works Association*. (Published in *Journal* of *NEWWA*, Vol. XLIV, No. 4, and reprinted as Publication from Harvard Engineering School, No. 58, 1930-31.)

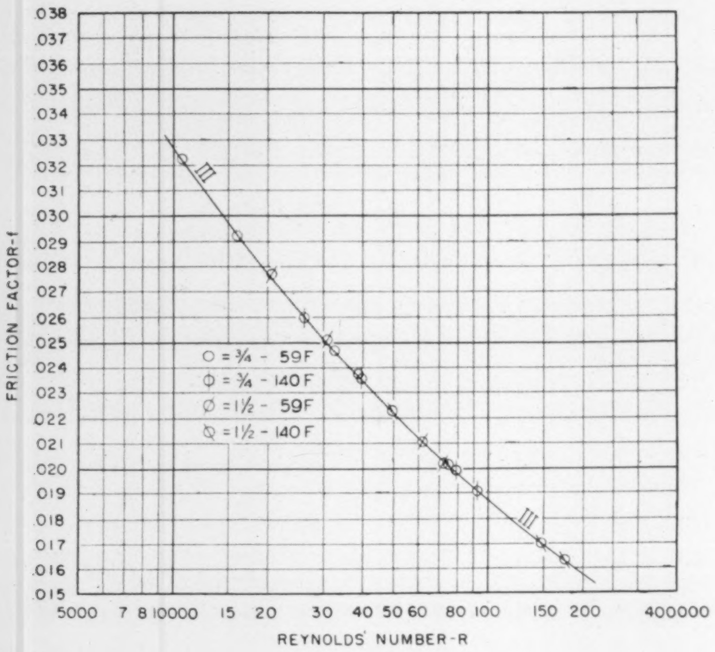

Fig. 3—The Fair-Whipple-Hsiao *f-R* line

Sixteen values calculated by means of formulae $h = 0.307\ d^{-1.246}\ v^{1.754}$ and $h = 0.247\ d^{-1.246}\ v^{1.754}$ for 59 F and 140 F water, respectively; and line representing the average *f-R* relation.

Table 3—Tests by G. M. Fair, M. C. Whipple, and C. Y. Hsiao—1930 and earlier

Nominal Diameter Inches	Actual Diameter Feet	Temperature F	Velocity fps	Friction Head Ft per 100 ft	*f*	*R*
¾	0.0676	59	2	2.9719	0.0323	11,000
¾	0.0676	59	3	6.0520	0.0293	16,400
¾	0.0676	59	6	20.413	0.0247	32,900
¾	0.0676	59	7	26.750	0.0238	38,400
1½	0.1273	59	2	1.3506	0.0277	20,600
1½	0.1273	59	3	2.7504	0.0251	30,900
1½	0.1273	59	6	9.2768	0.0211	61,900
1½	0.1273	59	7	12.157	0.0203	72,200
¾	0.0676	140	2	2.3911	0.0260	26,400
¾	0.0676	140	3	4.8692	0.0236	39,600
¾	0.0676	140	6	16.423	0.0199	79,200
¾	0.0676	140	7	21.522	0.0191	92,400
1½	0.1273	140	2	1.0866	0.0223	49,700
1½	0.1273	140	3	2.2129	0.0202	74,600
1½	0.1273	140	6	7.4638	0.0170	149,200
1½	0.1273	140	7	9.7810	0.0164	174,000

changes in hydraulic characteristics of pipes with service, but they include, necessarily, studies of the friction heads in new pipes. The authors concluded from their studies that for copper, red brass and admiralty metal pipes, 1½ in. and ¾ in. in diameter, the friction heads may be calculated by the formulae:

$h = 0.307\ d^{-1.246}\ v^{1.754}$ ft per 1000 ft and for 59 F water..................(3)

$h = 0.247\ d^{-1.246}\ v^{1.754}$ ft per 1000 ft of pipe and for 140 F water........(4)

if d is in feet and v in feet per second. In accordance with these equations, four friction heads were calculated for ¾ in. pipe and 59 F water; four for ¾ in. pipe and 140 F water; four for 1½ in. pipe and 59 F water; and four for 1½ in. pipe and 140 F water. Then, for each case the values of *f* and *R* were calculated. The results are shown in Table 3 and in Fig. 3.

In a study[5] conducted at the Agricultural and Mechanical College of Texas in 1931, friction heads were determined for 84 F water flowing in 1¼-in., 1-in., and ½-in. copper tubes 30 ft in length. The investigators concluded that, under these conditions, the friction heads could be calculated with sufficient accuracy by means of the formula:

$h = 1.16\ d^{-1.22}\ v^{1.7}$ milinches per foot for 84 F water..................(5)

if d is in inches and v, in feet per second. Equation (5) becomes:

$h = 0.319\ d^{-1.22}\ v^{1.7}$ ft per 1000 ft for 84 F water......................(6)

if d is in feet and v, in feet per second. Five friction heads were calculated by means of Equation (5) for each of the four pipe sizes and then, for each case, the values of *f* and *R* were calculated. The results are shown in Table 4 and in Fig. 4.

Comparing Fig. 4 with Fig. 3 and Equation (6) with Equation (4), it appears that the two equations are similar but that in Equation (4) the exponents of v and d were determined to a higher degree of accuracy than was the case in Equation (6). In Fig. 3 the 16 calculated values lie practically on an unbroken line; whereas, in Fig. 4 the five calculated values for the four pipe sizes lie on four distinct lines so that for any one value of *R* the extreme *f* values vary about 3 per cent from the mean. The explanation may be that, while Equation (5) is sufficiently accurate for ordinary design purposes, it is not sufficiently accurate for the study in hand.

The four *f-R* lines shown in Figs. 1, 2, 3, and 4 were transferred and reproduced in Fig. 5. It is evident from this figure:

1. that the Freeman *f-R* line extends over a wider range of *R* values than any of the other three lines.
2. that the Freeman *f-R* line shows values which differ only slightly from those shown by the other lines. For example, for an *R* value of 40,000, which is near the center of range of the other three *f-R* lines, the value shown by the Freeman line is only about 2 per cent lower than the average of the values shown by the other three.

It seems, therefore, that the Freeman *f-R* line should be used as the basis for friction head calculations for hydraulically *smooth* pipes, especially, also, since the Freeman values are based on tests of pipe ranging in size up to 4 in.; whereas, the other lines are based on tests of pipe ranging up in size only to 2 in.

Before deciding definitely on an *f-R* line for friction head calculations, it should be borne in mind:

1. that the Freeman tests were made with very long pipes. Of the 35 tests recorded in Table 2 and Fig. 2, 25 were longer than 100 ft.
2. that the friction head in a pipe is the result of the friction between the molecules of the water in the liquid

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[5] ASHVE Research Report No. 939—Loss of Head in Copper Pipe and Fittings, by F. E. Giesecke and W. H. Badgett. (ASHVE Transactions, Vol. 38, 1932.)

Table 4—Tests by F. E. Giesecke and W. H. Badgett—1931-32

Nominal Diameter Inches	Actual Diameter Feet	Temperature F	Velocity fps	Friction Head Ft per 100 ft		*R*
¾	0.0676	84	2	2.771	0.03015	15,240
¾	0.0676	84	3	5.521	0.02671	22,860
¾	0.0676	84	5	13.149	0.02290	38,100
¾	0.0676	84	6	17.937	0.02169	45,720
¾	0.0676	84	7	23.310	0.02071	53,340
1	0.0879	84	2	2.010	0.02845	19,820
1	0.0879	84	3	4.0052	0.02519	29,740
1	0.0879	84	5	9.545	0.02161	49,560
1	0.0879	84	6	13.013	0.02046	59,470
1	0.0879	84	7	16.912	0.01954	69,380
1¼	0.1076	84	2	1.57	0.02720	24,260
1¼	0.1076	84	3	3.13	0.02410	36,380
1¼	0.1076	84	5	7.46	0.02068	60,640
1¼	0.1076	84	6	10.17	0.01958	72,770
1¼	0.1076	84	7	13.22	0.01870	84,900
1½	0.1273	84	2	1.28	0.02623	28,700
1½	0.1273	84	3	2.55	0.02324	43,050
1½	0.1273	84	5	6.08	0.01994	71,750
1½	0.1273	84	6	8.29	0.01888	86,080
1½	0.1273	84	7	10.77	0.01802	100,420

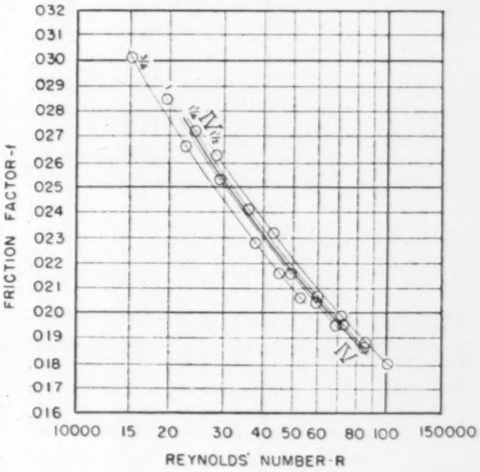

Fig. 4—The Giesecke-Badgett *f-R* line

Five values each for 1½ in., 1¼ in., 1 in., and ¾ in. copper tubes and 84 F water calculated by means of the formula $h = 1.16\ d^{-1.22}\ v^{1.7}$, with lines showing the *f-R* relation for the four pipe sizes individually and collectively within the range from $R = 22{,}000$ to $R = 88{,}000$, in which the experimental determinations were made.

stream. During turbulent flow—which exists for all velocities corresponding to *R* values above 3,000, approximately—a turbulence is produced in the liquid stream, possibly by the roughness of the inner pipe surface, and other factors. This turbulence attains a steady state in a long straight pipe and in turn produces a steady or constant *f* value. When the water flows through an elbow or valve, or any other similar obstruction, an additional turbulence is produced. This additional turbulence continues for a considerable distance beyond the obstruction in which it was produced and is superimposed on the turbulence which exists as the result of ordinary turbulent flow. Consequently, when water is flowing through a pipe fitting into a long straight pipe, the average friction head in that pipe becomes less and less as the pipe length becomes greater and greater. The friction factor will be less for a pipe 50 ft long than for a pipe 20 ft long.

Since, in ordinary hot water heating systems, the lengths of straight pipe runs between fittings are generally less than 12 ft and, since the Freeman results are based on much longer pipes, it is wise to add a certain percentage to the Freeman values if the resulting friction heads are to be used in the design of ordinary hot water heating systems.

For these reasons the heavy line in Fig. 5 was drawn to show *f* values 5 per cent higher than those shown by the Freeman line, and this line is suggested as the basis for friction head calculations for the design of hot water heating systems. Having such a line, it is easy to calculate the friction head for any desired condition if the *R* value for that condition is known. To find the *R* value it is necessary to know the kinematic viscosity of the water for the desired condition.

The author was unable to find a satisfactory table of kinematic viscosities in English units, so he prepared Table 5. The values in this table are based on data found in International Critical Tables and other reference books.

Having the *f-R* line of Fig. 5 and the kinematic viscosities of Table 5, the friction heads may be found for any condition. For example, for a 4 in. Type *M* copper tube, when water at 200 F is flowing in the pipe at a velocity of 8 fps, Reynolds' number (*R*) is found by multiplying the velocity of the water, in feet per second, by the internal diameter of the pipe, in feet, and dividing the resulting product by the kinematic viscosity in feet squared per second. In the present case *R* is 8 x 0.328 ÷ 0.00000343 or 765,000. For this

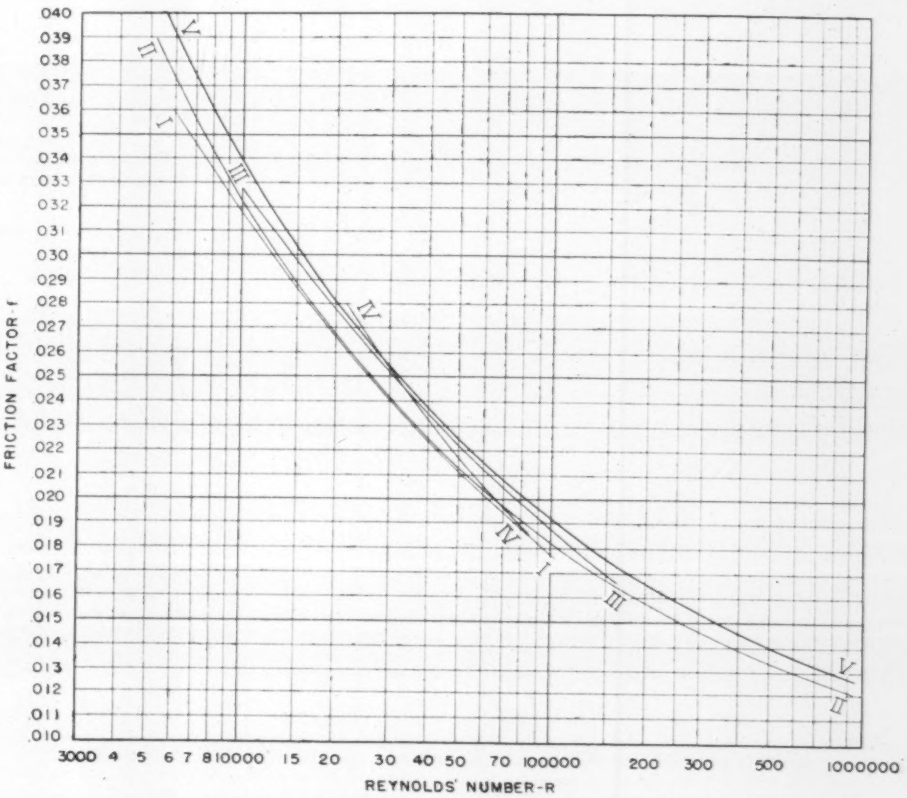

Fig. 5—A comparative presentation of four *f-R* lines of Figs. 1, 2, 3, and 4 and major line showing values of *f* which are 5 per cent higher than those shown by the Freeman *f-R* line

Table 5—Kinematic Viscosities of Water
(Feet squared divided by seconds)

F		F		F	
35	0.00001822	105	0.00000700	175	0.00000395
40	0.00001664	110	0.00000665	180	0.00000382
45	0.00001528	115	0.00000633	185	0.00000371
50	0.00001410	120	0.00000604	190	0.00000361
55	0.00001307	125	0.00000578	195	0.00000352
60	0.00001216	130	0.00000554	200	0.00000343
65	0.00001134	135	0.00000532	205	0.00000334
70	0.00001059	140	0.00000512	210	0.00000325
75	0.00000991	145	0.00000493	215	0.00000316
80	0.00000931	150	0.00000475	220	0.00000307
85	0.00000876	155	0.00000457	225	0.00000299
90	0.00000826	160	0.00000440	230	0.00000291
95	0.00000781	165	0.00000424	235	0.00000283
100	0.00000739	170	0.00000409	240	0.00000276

value of *R,* Fig. 5, *Curve V,* shows 0.0131 as the corresponding *f* value. The friction is then [Equation (1)]

$$\frac{0.0131}{0.328} \times \frac{64}{64.4} \text{ or } 0.003969 \text{ ft per}$$

foot or 476 mi per foot. If, in this case, the temperature of the water were reduced from 200 F to 140 F, the kinematic viscosity would be increased from 0.00000343 to 0.00000512; the value of *R* reduced from 765,000 to 513,000; the value of *f* increased from 0.0131 to 0.014; and the friction head increased from 476 mi per foot to 509 mi per foot, an increase of about 7 per cent.

To prepare a chart like that shown in Fig. 6 it is only necessary to assume a basic temperature, calculate the friction heads for 2 pipe sizes and for 2 or 3 velocities in those pipes, plot the resulting values to logarithmic coordinates, find the equations of the resulting lines, and combine these equations into one general equation from which the friction heads for the remaining pipe sizes can be determined. For example, if 140 F is selected as the basic temperature and friction heads calculated for velocities of 1, 3, and 8 fps in 1 in. and 6 in. pipes, the friction heads will be as shown in Table 6. After plotting these 6 values to logarithmic coordinates as indicated by circles in Fig. 6, the equations for the friction heads may be found to be:

For 1 in. pipe, $h = 61.833\ v^{1.7028}$ mi. (7)
For 6 in. pipe, $h = 7.346\ v^{1.8102}$ mi. . (9)

if v is in feet per second.

From these two equations, the general equation,

$$h = 66.07\ d^{-1.2398} v^{1.7613\ d^{0.01544}} \quad ..(10)$$

if v is in feet per second and d, in in., may be determined and the chart completed as shown in Fig. 6.

Although the values shown in this chart are not scientifically exact since the construction of the chart is based on a rectilinear *f-R* relation, they are sufficiently accurate to be used in the design of hot water heating systems and in all other ordinary hydraulic calculations relating to 140 F water flowing in copper tubes or other similar pipes. This is evident from Table 7, in which accurately calculated friction heads are compared with corresponding friction heads shown by the chart.

The reason for selecting 140 F

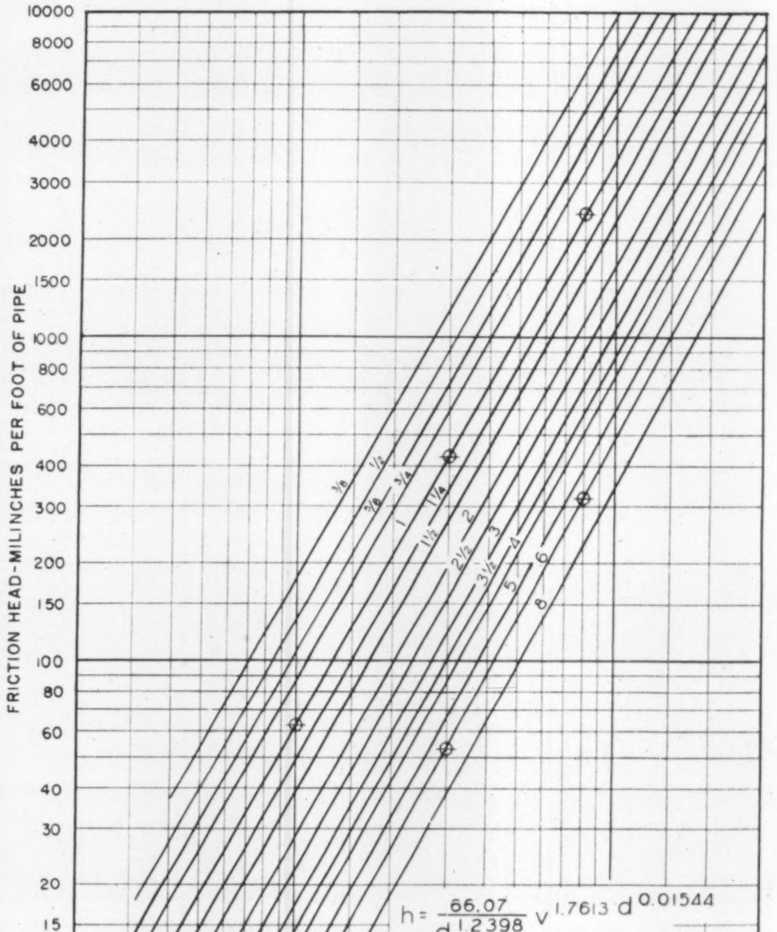

Fig. 6—Friction heads based on the major *f-R* line

Curve V of Fig. 5 for 140 F water flowing in Type *M* copper tubes. Whenever the temperature of the water differs materially from 140 F or when the water is flowing in iron or other similar pipes, the friction heads shown in this chart should not be used.

Table 6—Friction Heads Determined from *f-R* Line, Fig. 5, *Curve V*
(For Type M Copper Tubes and 140 F Water)

NOMINAL DIAMETER INCHES	ACTUAL DIAMETER INCHES	VELOCITY FPS	FRICTION HEAD MI PER FT	*f*	*R*
1	1.055	1	61.83	0.0292	17,200
1	1.055	3	422.69	0.0222	51,500
1	1.055	8	2421.72	0.0179	137,300
6	5.881	1	7.35	0.0193	95,700
6	5.881	3	52.82	0.0154	287,200
6	5.881	8	316.82	0.0130	765,800

Table 7

NOMINAL DIAMETER INCHES	ACTUAL DIAMETER INCHES	VELOCITY FPS	*R*	*f*	FRICTION HEAD CALCULATED	FRICTION HEAD FROM CHART
⅜	0.450	0.8	5,860	0.0396	126	123
2	2.009	4	131,080	0.0181	322	327
8	7.785	7	887,030	0.0128	180	178

as the basic temperature is that it is believed to be the temperature at which hot water heating systems operate during the larger part of the heating season. If other temperatures are preferred as basic temperatures, corresponding charts can be easily prepared.

Acknowledgment

The author is indebted to Cadets J. F. Gordon, A. S. Ware, Jr., and R. F. Worth for assistance in preparing calculations and drawings and to Mrs. Barbara Redding for assistance in preparing manuscript and tables.

NEW VAPOR TRANSMISSION UNITS PROPOSED

During the past few years there has been a considerable amount of experimental work conducted in the field of vapor transmission, particularly in connection with vapor flow through building materials. However, practically all of the investigators and authors have used different units for expressing the results obtained. This has resulted in a great deal of confusion and has made it difficult to compare the results of the various investigators. Furthermore, in many cases it has been the practice to use both metric and English units rather than to use one or the other.

With the object of overcoming some possible confusion resulting from this practice and assisting in establishing uniform units for expressing vapor flow, the Technical Committee of the Insulation Board Institute has made a study of this subject and has developed what it considers to be desirable English system units for this purpose, as follows:

Grains per square foot per hour per inch mercury.

This method of stating the results not only employs English units throughout, but the units involved are of a convenient magnitude. For example, it will be found that the desirable maximum rate of vapor flow for the preclusion of wall condensation as expressed by at least two prominent investigators will give values approximating unity (1.0) when the results are stated in terms of these units. The rate of vapor flow through the majority of materials will range between 0.5 and about 100 on this basis.

Another advantage, according to the Insulation Board Institute Technical Committee, is that the use of *per inch mercury* for expressing pressure difference eliminates the use of two area quantities such as for example, is the case where the vapor flow is expressed in *grains per square foot per hour per pound per square inch.*

The Insulation Board Institute Technical Committee is suggesting to all concerned with this subject that where the English system is employed these units be adopted by experimenters and authors in the future in reporting test results and in writing on this subject.

Vapor Transmission Conversion Factors
To convert to grains per square foot per hour per inch mercury, multiply by the factors in this table.

AMOUNT OF VAPOR	TIME	PRESSURE DIFFERENTIAL		
		PER MM HG	PER INCH HG	PER LB PER SQ IN.
A	B	C	D	E
Grams per square foot............	Per hour	392.0	15.43	7.57
Grams per square meter..........	Per hour	36.3	1.43	0.702
Grains per square foot............	Per hour	25.4	x	0.491
Grams per square foot............	Per 24 hours	16.35	0.643	0.315
Grams per square meter..........	Per 24 hours	1.52	0.0597	0.0293
Grains per square foot............	Per 24 hours	1.06	0.0417	0.0205

C. S. BECKER JOINS WPB

C. S. Becker, Manager of the Milwaukee Office of the American Blower Corp., manufacturers of ventilating equipment, has resigned from that position effective October 1, to become associated with the Heating and Plumbing Branch of the War Production Board. His new headquarters will be in Washington, D.C. Mr. Becker was a member of the Tau Beta Pi and Triangle Fraternities at Marquette University from which he graduated in 1930 with a degree of Mechanical Engineer. In addition to his regular work, Mr. Becker was serving as Secretary of the Wisconsin Chapter of the AMERICAN SOCIETY OF HEATING and VENTILATING ENGINEERS.

He is being succeeded as Manager of the American Blower Corp. by M. W. Bishop who has been located in the Chicago office of that concern for the last eleven years, and as Secretary of the Wisconsin chapter by I. J. Haus.

The Performance of Side Outlets on Horizontal Ducts

By D. W. Nelson* and G. E. Smedberg,** Madison, Wis.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the University of Wisconsin

SUMMARY—Actual measurements are given for the air distribution pattern leaving the side outlets of a horizontal duct. Data for three sizes of outlets are reported with various types of corrective devices applied. Complete specifications are outlined for an air flow analyzer, which was used to investigate the flow pattern from the various outlets. The results indicated that the use of curved deflectors showed the best flow characteristics from the standpoint of uniformity of flow, negative area, and static pressure

Introduction

Horizontal ducts with side outlets as used in heating and ventilating installations must have the flow of air controlled to give the desired effect. The performance of three sizes of side outlets on a horizontal duct equipped with three types of corrective devices and various extension lengths has been investigated. The results obtained should be of assistance in designing effective outlets to the end that distribution in rooms may be satisfactory.

Description of Apparatus

The test set-up used is shown in Figs. 1 and 2. The radial fan was connected by means of a suitable reducing section to an 18 in. x 18 in. duct 12 ft long. At the end of this 12-ft section a 7.55 in. nozzle was inserted. Downstream of the nozzle was a 7 ft length of 18 in. x 18 in. duct, in which were located adjustable and stationary static and total pressure tubes. The adjustable tubes were used in the calibration of the nozzle and the stationary tubes for setting the velocity during the various tests. This 7 ft length of 18 in. x 18 in. duct was followed by a 6 in. x 20 in. section of duct 15 ft long. Two openings were located in the side of this 6 in. x 20 in., and a third opening was located at the end of the duct. These openings were designed to permit the insertion of plates with the various sizes and types of outlets. The plates with the outlets had 1 in. extensions and were designed to permit the attachment of 2 in., 4 in., 6 in., and 12 in. long extension pieces resulting in total extension lengths of 1 in., 3 in., 5 in., 7 in., and 13 in. Three sizes of outlets were studied; namely, 3 in. x 10 in., 4 in. x 9 in., and 6 in. x 6 in. The static pressures in the system were measured by means of piezo meter rings and static pressure tubes at the points numbered 1 to 12 in Fig. 2. The dry- and wet-bulb temperatures of the air flowing in the duct were measured by means of suitable thermometers placed at the entrance to the 6 in. x 20 in. duct.

The volume of air flowing in the system was controlled by a damper and atmospheric relief door in the duct adjacent to the fan outlet and an adjustable cone on the inlet of the fan.

Readings were taken using a deflecting vane anemometer with a duct jet at three stations in addition to flow measurements at the nozzle station. These stations were located before outlet No. 1, between outlet Nos. 1 and 2 and between outlet Nos. 2 and 3.

The three outlets were divided into 24 equal areas and the velocity at the center of each of these areas was measured. Angles of flow were measured at the center of each area for the middle row on each outlet.

Directive devices as shown in the lower right hand corner of Fig. 2 were designed to permit their convenient insertion at each of the side outlets. These devices were not used at outlet No. 3 during the various tests, but the same length of extension was used on this outlet as on Nos. 1 and 2 in each test.

The test set-up was arranged so that all the static pressures could be read from a single point, and cables were arranged permitting the operation of the control dampers from the same location. Pressures were measured by differential or inclined manometers, depending on the range of the static pressures to be measured.

An air flow analyzer shown in Fig. 3 was used to study the flow pattern from the outlets. It has two glass plates spaced ¼ in. apart between which the model is placed, and between which the air flows, hence, it is limited to two dimensional flow. The model duct or shape to be studied may be conveniently constructed from rubber strips or rub-

Fig. 1—Photograph of apparatus

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*Associate Professor of Mechanical Engineering, University of Wisconsin. Member of ASHVE.

**Research Fellow, University of Wisconsin, 1940-1942.

For presentation at the 49th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cincinnati, Ohio, January 1943.

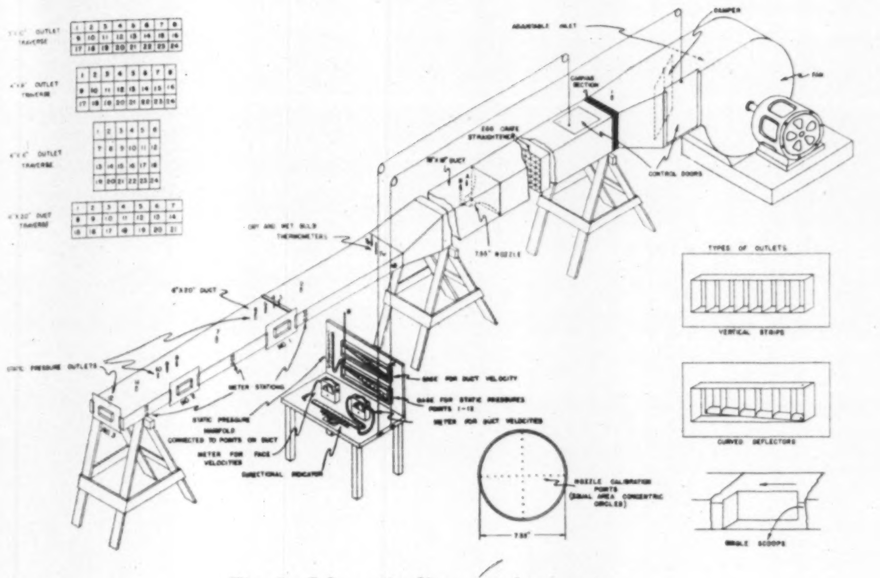
OUTLET TRAVERSE
4"x8" OUTLET TRAVERSE
6"x6" OUTLET TRAVERSE
6"x20" DUCT TRAVERSE
ADJUSTABLE INLET
DAMPER
FAN
EGG CRATE STRAIGHTENER
CONTROL DOORS
7.55" NOZZLE
DRY AND WET BULB THERMOMETERS
6"x20" DUCT
STATIC PRESSURE OUTLETS
METER STATIONS
STATIC PRESSURE MANIFOLD CONNECTED TO POINTS ON DUCT
METER FOR FACE VELOCITIES
DIRECTIONAL INDICATOR
GAGE FOR DUCT VELOCITY
GAGE FOR STATIC PRESSURES POINTS 1-12
METER FOR DUCT VELOCITIES
NOZZLE CALIBRATION POINTS (EQUAL AREA CONCENTRIC CIRCLES)
7.55"
TYPES OF OUTLETS
VERTICAL STRIPS
CURVED DEFLECTORS
SINGLE SCOOPS

Fig. 2—Schematic diagram of apparatus

ber sections. A slotted trough placed at the upstream end of the device is filled with hydrochloric acid back of which is an open container of ammonium hydroxide. The chemical combination of the two vapors forms ammonium chloride particles, which are carried in streamlines through the apparatus by the air stream. The fan can be used at either end and when on the upstream side it acts as a supply fan, blowing the streamlines over the model. When used in this manner the plenum chamber on the upstream side is used. This chamber consists of a baffle arrangement which results in an even flow across the entire width of the apparatus. When used in this manner a vent strip is removed from the side of the apparatus allowing the streamlines to be projected outward from the opening in the duct work into the atmosphere. This method was used for the study of air flow in ducts since it was possible to discharge the smoke streamlines directly into the atmosphere which simulates actual duct operation. If the fan is placed on the downstream side of the apparatus, it acts as an exhaust fan drawing the streamlines over the model. Operating in this manner the vent strip on the side is closed and the streamlines of smoke are drawn straight through the apparatus and discharged to the outside by suitable piping. This method of operation is used when studying the flow over various shaped objects such as wing sections and cross sections of condenser or boiler tubes, and reference to the literature shows that it has been used considerably in aeronautic studies.

The flow of air over various objects can be visually studied by means of this device, and a great deal of valuable knowledge thereby obtained. However, by recording the flow photographically the results can be studied at leisure and are of value for illustrating various points of theory. When the photographs for this paper were taken, the camera was mounted directly over the glass plates on a rigid support. The built-in lighting arrangement consisting of twenty-four 60-watt Mazda bulbs furnished sufficient illumination to allow a shutter speed of one-fiftieth to one-hundredth of a second with an aperture of *f*2.8. These speeds are based on the use of a film with a Weston tungsten rating of 64.

Procedure

Duct velocities of 200, 500, 800, 1100, and 1400 fpm in the 6 in. x 20 in. section were maintained in the series of tests on various sizes and types of outlets and with various extension lengths. Tests were run at each of these velocities for the three sizes of outlets with 1 in. and 13 in. extensions. Only 1400 fpm was used for the intermediate lengths since preliminary tests indicated similar flow patterns were obtained as at lower velocities. These series of tests were carried out on the plain outlets, single scoop, vertical strip and curved deflectors in position in the outlets.

Calculations were made, previous to each test, of the correct gage setting at the nozzle corresponding to the velocity desired in the duct before outlet No. 1. In a test of each outlet combination, the direction and velocity at the center of each of twenty-four equal areas on each of the three outlet faces were determined by means of a deflecting vane anemometer used with a spot reading tip and a vane type angle measuring device termed a directional indicator.[1] In addition, the static pressures were read at the 12 points of the system shown in Fig. 2. The

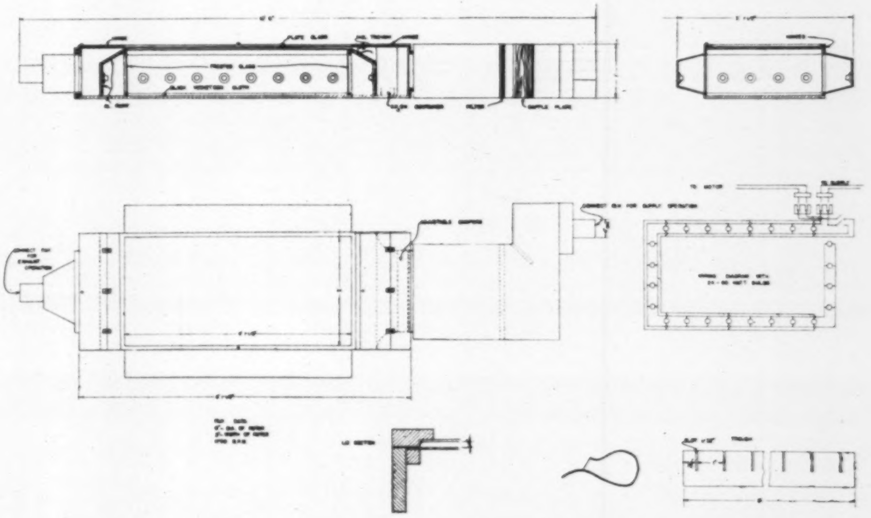

Fig. 3—Drawing of air flow analyzer

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[1] The Performance of Stack Heads Equipped with Grilles, by D. W. Nelson, D. H. Lamb, and G. E. Smedberg. See Fig. 2 (ASHVE Journal Section, *Heating, Piping and Air Conditioning*, January, 1942.)

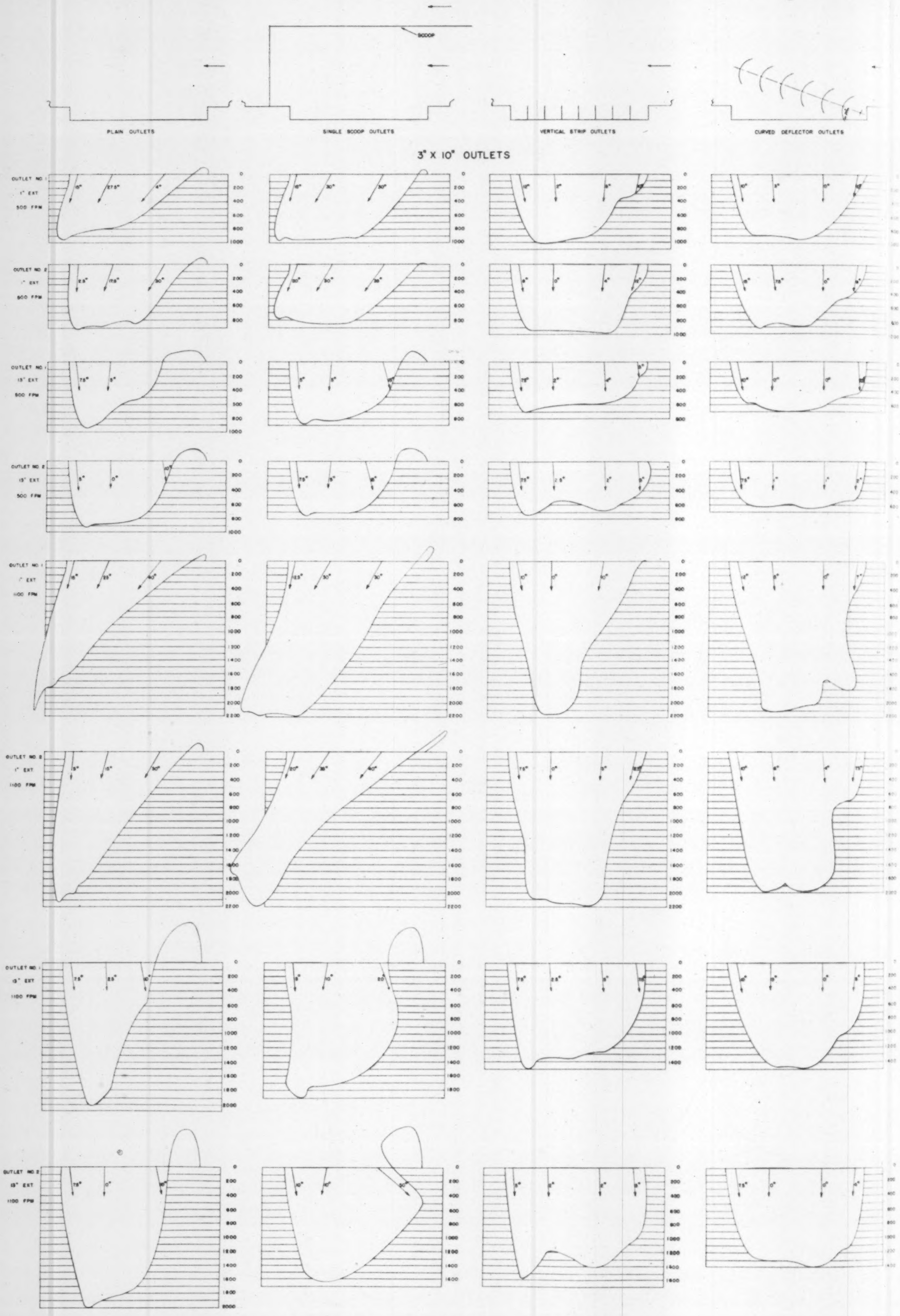

Fig. 4—Flow patterns for 3-in. x 10-in. outlet

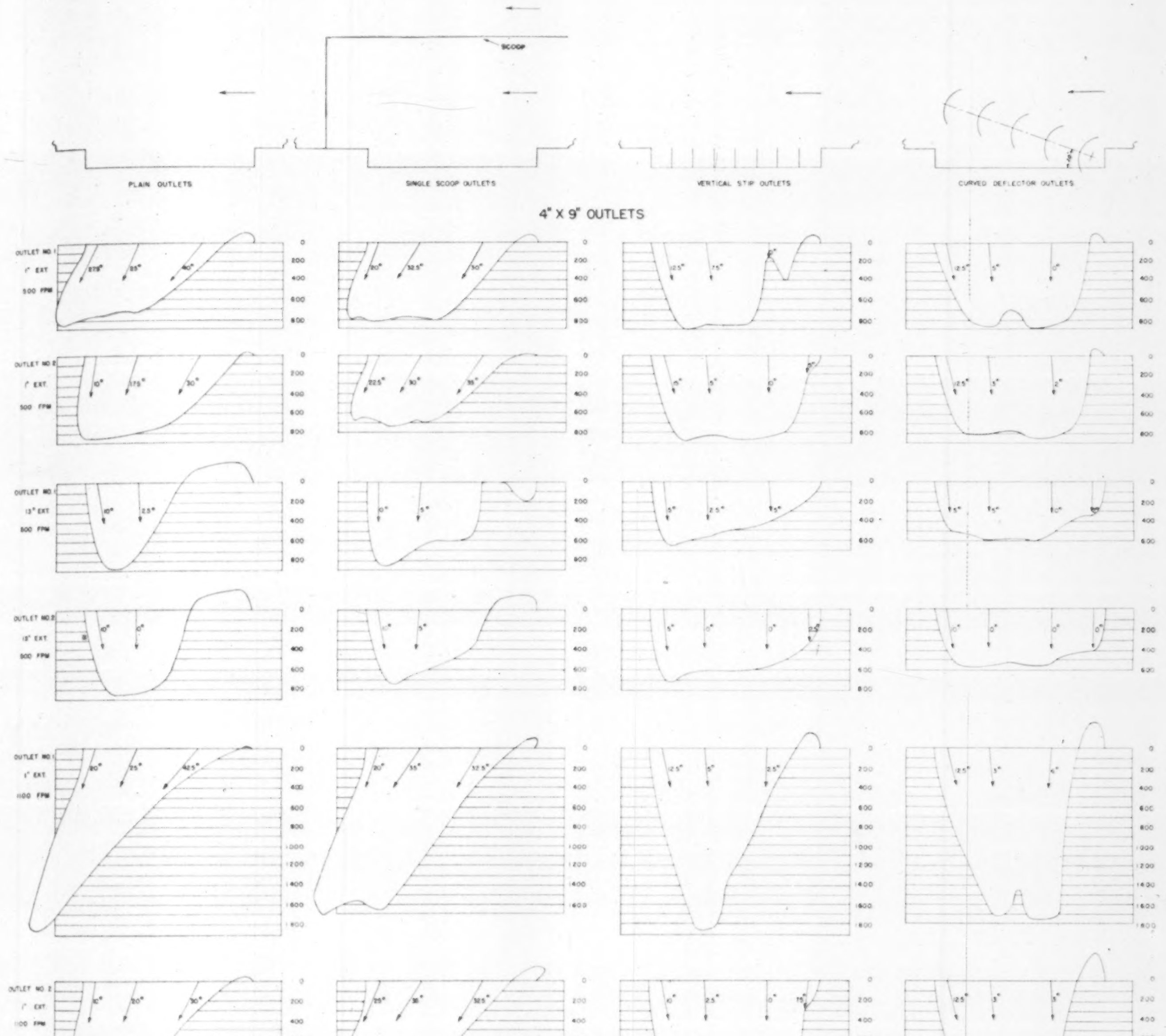

Fig. 5—Flow patterns for 4-in. x 9-in. outlet

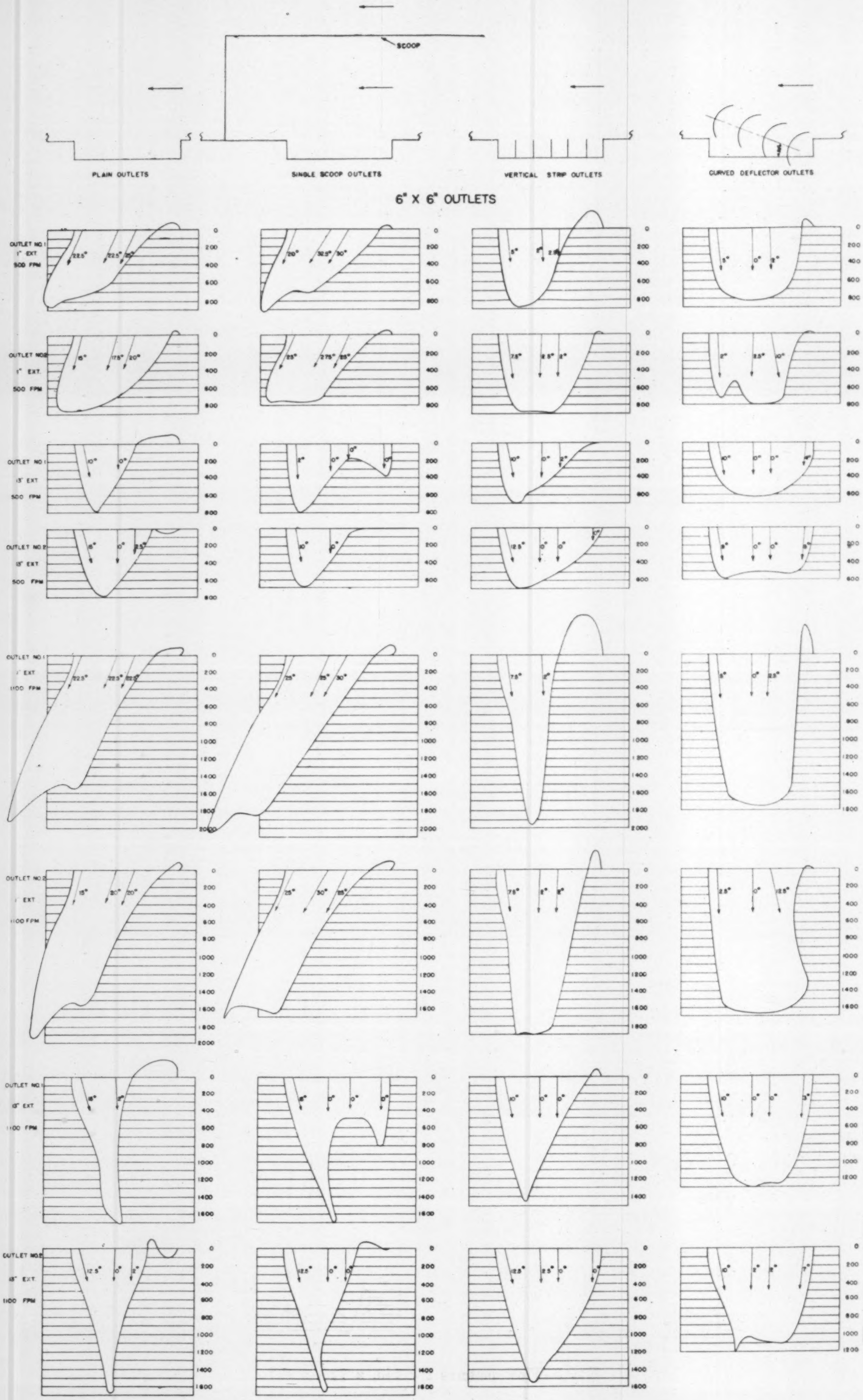

Fig. 6—Flow patterns for 6-in. x 6-in. outlet

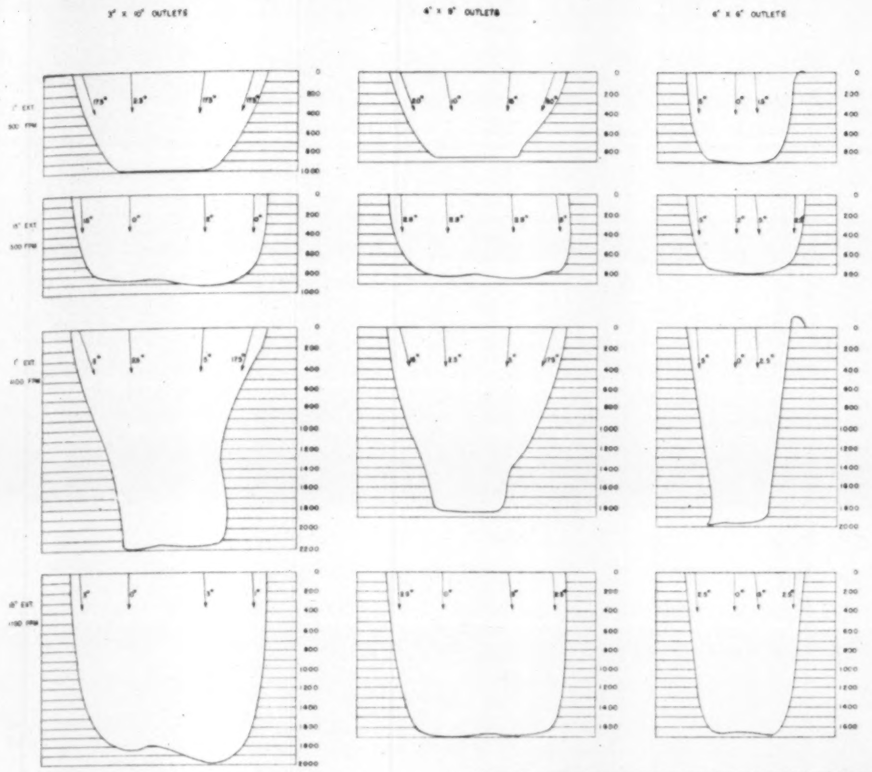

Fig. 7—Flow patterns for outlet No. 3

center velocity at the three measuring stations, where the deflecting vane anemometer was used with a duct jet, was also recorded during each test.

Summary of Results

Figs. 4, 5, and 6 show the flow patterns at 500 and 1100 fpm for the three sizes of outlets with the various directive devices, and with 1 in. and 13 in. extension lengths. With the plain outlet and 1 in. extension, a very large angle of discharge was obtained. The addition of extensions reduced this angle. The same general flow patterns prevail when single scoops were used before the outlets and devices of this sort appear to have no particular value. When used on 3 in. x 10 in. outlets, the vertical strip and curved deflector devices eliminated the negative flow area completely and resulted in an air discharge approximately perpendicular to the face of the outlet. Almost the same results were found on the 4 in. x 9 in. and the 6 in. x 6 in. outlets, except that a small area of negative flow remained when these two directive devices were used with a 1 in. extension on the outlets. However, even this small negative area was removed when longer extensions were used.

The flow characteristics of outlet No. 3 are shown in Fig. 7. This outlet was without any directive device during all the tests, but had the same length of extension as on the side outlets in any particular test. The flow from the outlet at the end of the duct was reasonably uniform for each of the three outlet sizes tested. Directive devices on an outlet at the end of a main duct or branch are not necessary. In cases where it is desirable that the air stream be directed, the use of vertical strips set at the proper angles would seem desirable.

Photographs taken in the air flow analyzer for three lengths of extension operated in conjunction with plain, single scoop, vertical strip and

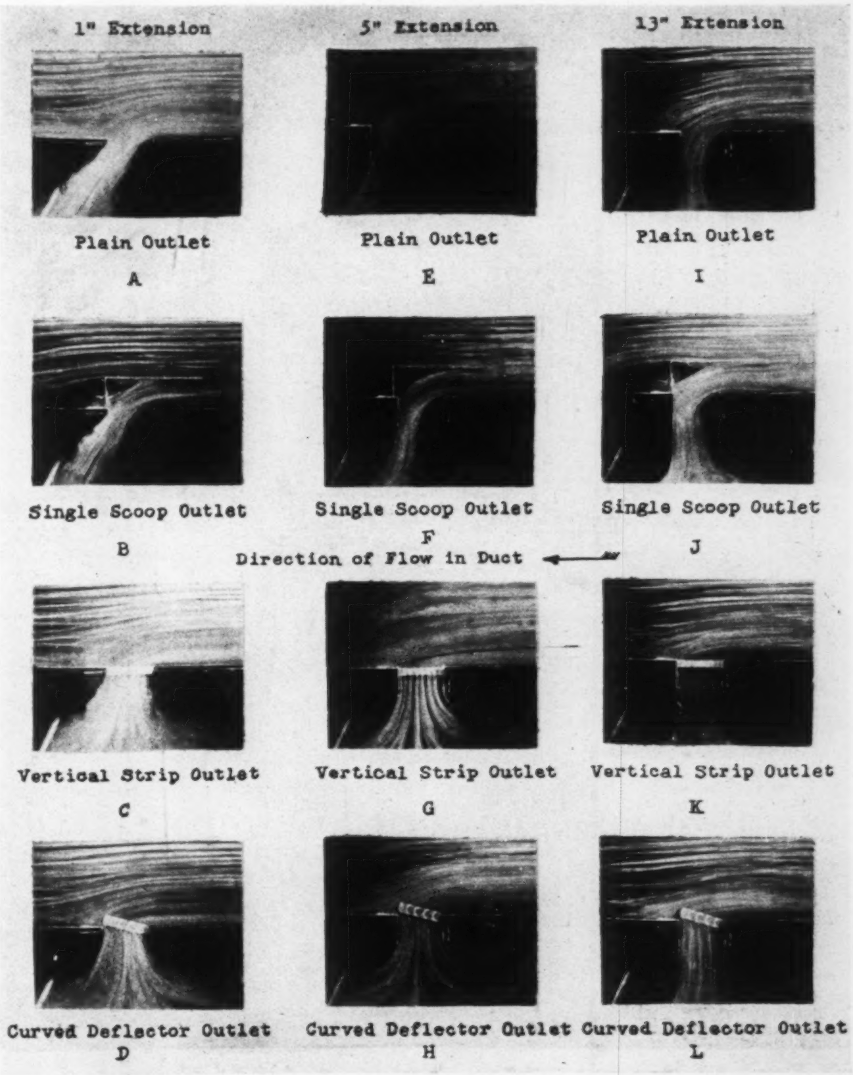
1" Extension
5" Extension
13" Extension
Plain Outlet
A
Plain Outlet
E
Plain Outlet
I
Single Scoop Outlet
B
Single Scoop Outlet
F
Single Scoop Outlet
J
Direction of Flow in Duct
Vertical Strip Outlet
C
Vertical Strip Outlet
G
Vertical Strip Outlet
K
Curved Deflector Outlet
D
Curved Deflector Outlet
H
Curved Deflector Outlet
L

Fig. 8—Air flow photographs with various types of outlets and extension lengths (6-in. x 20-in. duct and 4-in. x 9-in. outlets)

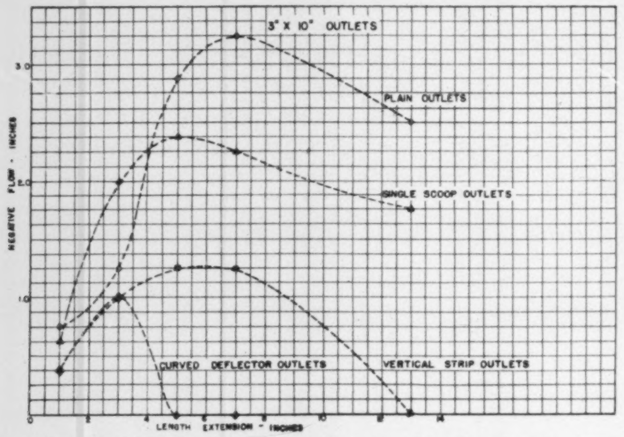

Fig. 9—Negative flow plotted against length extension

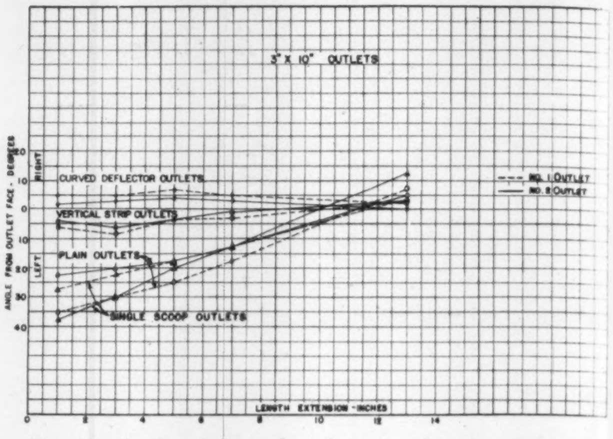

Fig. 10—Angle of flow plotted against length extension

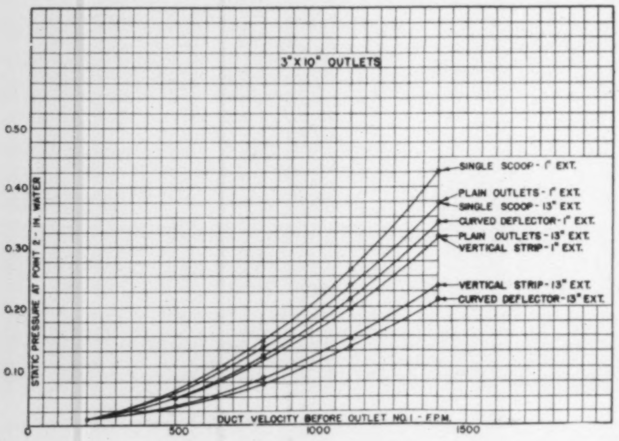

Fig. 11—Static pressure at point No. 2 plotted against duct velocity before outlet No. 1

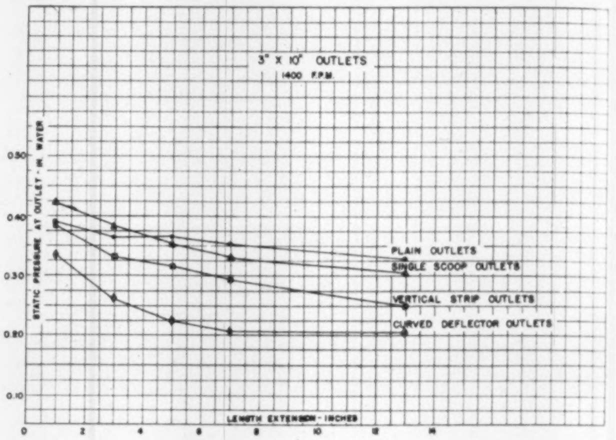

Fig. 12—Static pressure at outlet No. 1 plotted against length extension

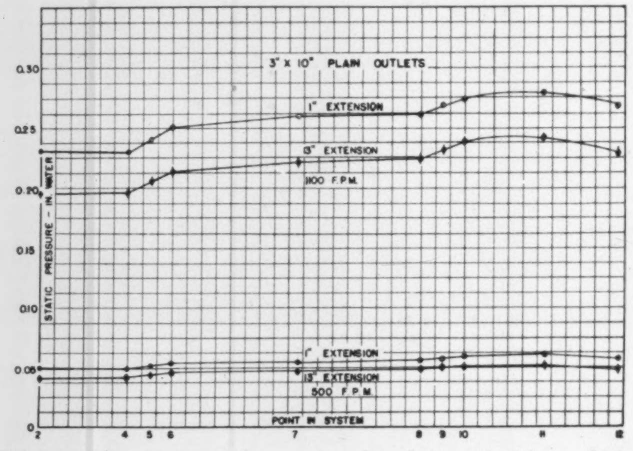

Fig. 13—Change in static pressure for duct with 3-in. x 10-in. plain outlets

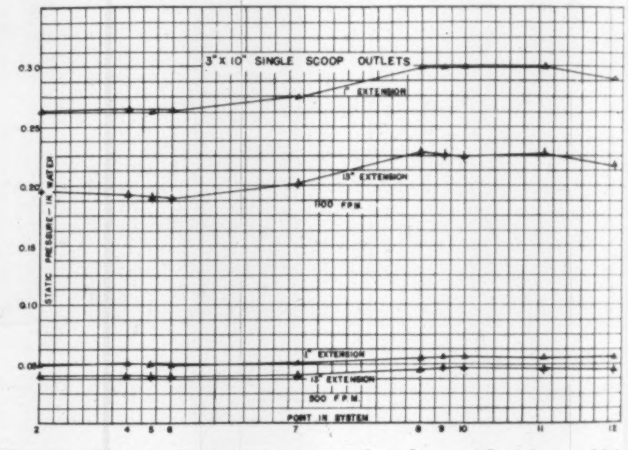

Fig. 14—Change in static pressure for duct with 3-in. x 10-in. single scoop outlets

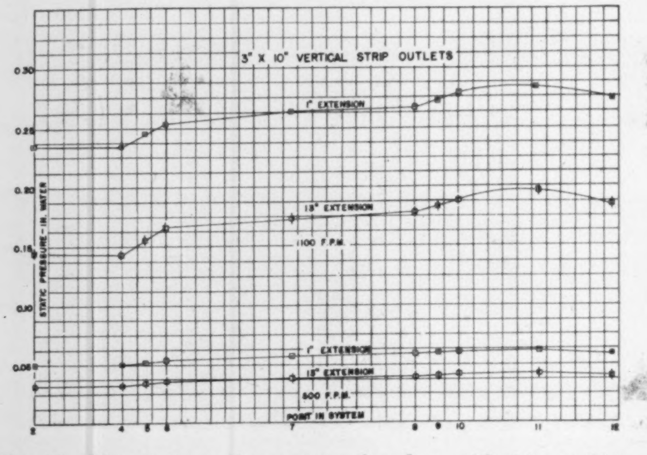

Fig. 15—Change in static pressure for duct with 3-in. x 10-in. vertical strip outlets

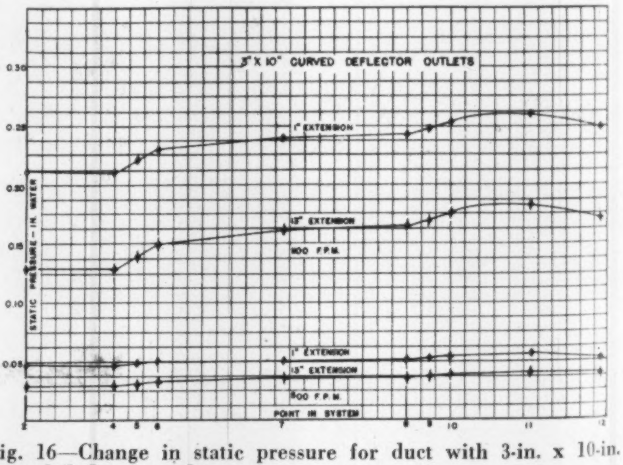

Fig. 16—Change in static pressure for duct with 3-in. x 10-in. curved deflector outlets

curved deflector types of outlets are shown in Fig. 8. The size of the duct used in the analyzer was one-half that of the duct used in the test and the outlet corresponded to 4 in. x 9 in. Fig. 8*a* shows the very pronounced angle of the air stream from the plain outlet with a 1 in. extension, and Fig. 8*b* shows that this condition is not corrected appreciably by use of a single scoop inside of the duct. Figs. 8*c* and 8*d* show the corrective effect of the vertical strips and curved deflectors.

The large negative flow area with a 5 in. extension is shown in Figs. 8*e* and 8*f*. The vertical strips and curved deflectors eliminated the negative flow as shown in Figs. 8*g* and 8. The air envelope on the up- 8*i* and 8*j* that an extension of 13 in. on the plain outlets was sufficiently long to eliminate the negative flow area. The curved deflector and vertical strip outlets gave the same results with this length extension as they did with the other lengths of extensions.

The amount of negative flow in inches is plotted as a linear distance from the upstream edge in Fig. 9 for various lengths of extensions for the 3 in. x 10 in. side outlets. The results with 4 in. x 9 in. and 6 in. x 6 in. outlets were similar. The plain outlets and the outlets with single scoops showed an increase and then a decrease in the negative flow area as the extension length was increased. This behavior is shown by the flow pictures in Fig. 8. The air envelope on the upstream side of the extension has a pronounced curvature which results in the complete spread across the face as the extension is increased in length.

In Fig. 10 the angle from the outlet face is plotted against the length of extension in inches for the 3 in. x 10 in. outlets. The angle of flow for the curved deflector and the vertical strip devices placed at the outlet is practically perpendicular to the outlet face for all lengths of extension. However, some further correction takes place in these cases with the use of the longer extension, but not as pronounced as with the plain and single scoop outlets. In the case of the latter, the increase in the extension length from 1 in. to 10 in. caused the angle of flow for outlet No. 2 to change from 38 deg to perpendicular to the outlet.

The static pressure at point No. 2 is plotted against the duct velocity before outlet No. 1 in Fig. 11. The static pressure at this point is highest when the single scoop device is used before the outlet. This is reduced somewhat as the length of extensions is increased, but still is as high as for the plain outlets without long extensions. The curved deflectors in conjunction with the 13 in. extension gave the lowest static pressure in every case. At a velocity of 200 fpm the change of static pressure was very small, but at 1400 fpm duct velocity the reduction secured by the use of the curved deflectors with 13 in. extension was 50 per cent greater than that obtained with a single scoop.

In Fig. 12 the static pressure at outlet No. 1 is plotted against the length of extension. The length of extension has an appreciable effect on the static pressure in the center of the duct in line with the outlet face. The reduction by the use of extensions is more pronounced with the vertical strips and curved deflectors than with the plain outlet. This reduction in static pressure can be expected to decrease beyond a certain length of extension due to additional frictional resistance in the longer extension.

In Figs. 13, 14, 15, and 16 the variation of static pressure at twelve different points of the system equipped with the 3 in. x 10 in. outlet and various directive devices is plotted for velocities of 500 and 1100 fpm. In each case, except for the single scoop device, definite static pressure regain across the face is indicated. With the single scoop arrangement accurate readings of the static pressures were difficult to obtain across the face of the outlet in the duct. This was due to the single scoop being inserted at this point. However, a regain takes place as is evidenced by the increase of static pressure from point 2 to 7, and to 11 in Fig. 14. The static pressure regain shown in Figs. 13 to 16 is of the same nature as that obtained in transition sections with gradual expansion. In the case of transition sections, the size of the duct is increased with a resultant decrease in velocity. This velocity decrease is accompanied by a conversion of the velocity head to static head with a resultant static pressure increase. In the case of the horizontal duct used in the present investigation, the duct size was held constant but the flow of air from the side outlets caused the reduction in the velocity past the outlet. There was as a result, an increase in the static pressure in the system past each outlet.

Conclusions

The use of curved deflectors resulted in the best flow characteristics from the standpoint of uniformity of flow, negative area, and static pressure. The vertical strips were not quite as effective as the curved deflectors. The latter have an advantage in that they can be placed at any angle to get the angle of flow desired. The single scoops do not correct the flow pattern over that of the plain outlet, and they do not appear to be desirable in duct installations. Several of these placed parallel to each other at the outlet face would possibly result in improved performance, but they would be less effective and more complicated than vertical strips. The use of volume dampers and grilles was not covered in the present investigation.

Acknowledgments

The tests on which this paper are based extended over a period of approximately two years. Acknowledgment is due to the following students in Mechanical Engineering for their valued assistance in the work: A. C. Holmes, C. L. Kaiser, W. H. Rowe, and H. L. Thies.

STEAM BOILERS NEEDED

One branch of the Army has estimated that 5700 boilers will be needed by April 30, 1943, according to the War Production Board, in addition to the demand by vital war industries for such equipment. All types of boilers in sizes from 35 hp and up, are required with operating pressures from 15 lb per square inch.

Anyone having boilers of this description available is urged to communicate with J. B. Joyner, Surplus Used Equipment Section, Conservation Division, War Production Board, 1100 H St., N.W., Washington, D.C.

Attic Temperatures, Ventilation and Heat Losses

By Paul D. Close*, Chicago, Ill.

SUMMARY—Two bases of calculation are presented for computing the heat loss through a roof section of a residence. Data are given on the use of combined roof and ceiling coefficients. A method is outlined for estimating the attic temperature with or without ventilation, or venting, and the number of air changes required to preclude the possibility of attic condensation. Actual attic temperature data obtained under operating conditions in a residence are reported for checking the theoretical assumptions.

HEAT FLOWS through top floor ceilings into unheated attics and thence through the roof and vertical wall areas to the outside. There are two methods by which this heat loss may be estimated: (1) using the combined roof and ceiling heat transmission coefficient, the inside-outside temperature difference and the roof or the ceiling area (depending on the basis of derivation of the combined coefficient), or (2) by estimating the attic temperature and using the ceiling coefficient and the area and the temperature difference through the ceiling. This paper includes a discussion of these two bases of calculation, data on combined roof and ceiling coefficients and a mathematical analysis of attic temperatures and the effect thereon of various amounts of ventilation.

Combined Coefficients

If the roof and ceiling are parallel, such as is the case with flat roofs, the respective resistances of the roof and ceiling are directly additive and the combined coefficients of these two components may readily be calculated. Heat transmission tables (such as those in THE ASHVE GUIDE) generally include combined roof and ceiling coefficients for flat roofs and parallel ceilings of various types of construction. Fig. 1 gives the combined coefficients for parallel roof and ceiling surfaces such as flat roofs with suspended or furred ceilings. A flat roof having a coefficient of 0.3 and a ceiling having a coefficient of 0.1, for example, will result in a combined coefficient of 0.075.

If the roof is pitched and the ceiling is not parallel therewith, the combined coefficient must be determined by a special formula which takes account of the angle of the roof pitch and is given in terms of either the roof or the ceiling surface, depending on the basis of derivation of the formula. The formula for the combined roof and ceiling coefficient for non-parallel surfaces expressed in terms of the *ceiling area* is as follows:

$$U = \frac{U_r \times U_c}{\frac{U_r + U_c}{n}} \ldots\ldots\ldots\ldots\ldots (1)$$

where

U_r = roof coefficient
U_c = ceiling coefficient
n = roof pitch or the ratio of the roof to the ceiling area (Table 1)

Fig. 2 shows the graphical relationship between the roof and ceiling coefficients and the combined coefficient for a one third pitch roof for which the value of n is 1.2. This roof pitch can be used with reasonable accuracy for all pitched roofs except extremely steep roofs. As an example of the use of Fig. 2, the combined coefficient of a roof having a coefficient of 0.2 and a ceiling having a coefficient of 0.4, based on a ⅓ pitch, is 0.15. Knowing the individual roof and ceiling coefficients, which can be obtained from standard reference books, the combined ccefficients can be more conveniently obtained from Fig. 2, than by means of Equation (1).

Fig. 2 and Equation (1) apply mainly to hip and double hip roofs (Fig. 3) and do not make allowance for vertical wall areas, dormers or ventilation. However, Equation (1) could be used with reasonable accuracy for gable roofs by including the gable wall area with the roof area in arriving at the value of n, and using a roof coefficient weighted according to the roof and gable wall coefficients and the respective roof and gable wall areas. A few trial calculations will show that even this correction would not be necessary in most cases where the roof is not steep, and the results would be sufficiently accurate for most heat loss calculations if the gable ends were neglected and the combined coefficient applied as it would be if the roof were of hip or double hip construction, using the ceiling area, the combined coefficient based on the ceiling area and the inside-outside temperature difference.

Estimating Attic Temperatures

In certain cases it is preferable to estimate the attic temperature and to calculate the heat loss by the second method referred to at the beginning of this article. The formula for estimating the attic temperature is as follows:

$$t_a = \frac{A_c U_c t_i + t_o(A_r U_r + A_w U_w + A_g U_g)}{A_r U_r + A_w U_w + A_g U_g + A_c U_c} \ldots (2)$$

where

t_a = attic temperature, degrees Fahrenheit
t_i = inside temperature near top floor ceiling, degrees Fahrenheit
t_o = outside temperature, degrees Fahrenheit
A_c = area of ceiling, square feet
A_r = area of roof, square feet
A_w = area of net vertical wall surface, square feet
A_g = area of glass, square feet
U_c = coefficient of transmission of ceiling
U_r = coefficient of transmission of roof
U_w = coefficient of transmission of vertical wall surface
U_g = coefficient of transmission of glass.

The temperature in an unheated attic on a cloudy day (no sunshine) will, of course, range between the inside air temperature in the rooms below and the outside temperature. If the attic is tight so that there is no air leakage or ventilation, the attic temperature will be a function of the ceiling, roof and vertical wall coefficients and areas. Fig. 4 gives the attic temperatures based on

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*Technical Secretary, Insulation Board Institute, MEMBER of ASHVE.

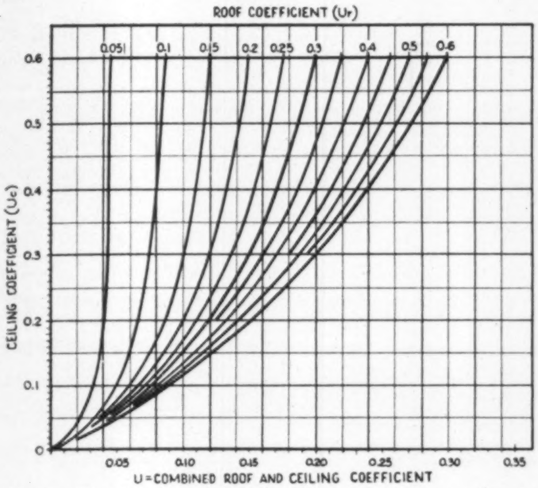

Fig. 1—Combined coefficients of transmission for parallel roof and ceiling surfaces (flat roofs)

Note: The values of U_r and U_c used with this chart should be based on one surface coefficient having a value equal to twice the air space conductance of 1.10, or 2.2, instead of the usual surface coefficient of 1.65. This will give the same result as using a conductance of 1.10 for the air space. The values of U derived from this chart should be used with the ceiling area. (For a further explanation, see ASHVE Guide 1942 beginning at the bottom of page 111).

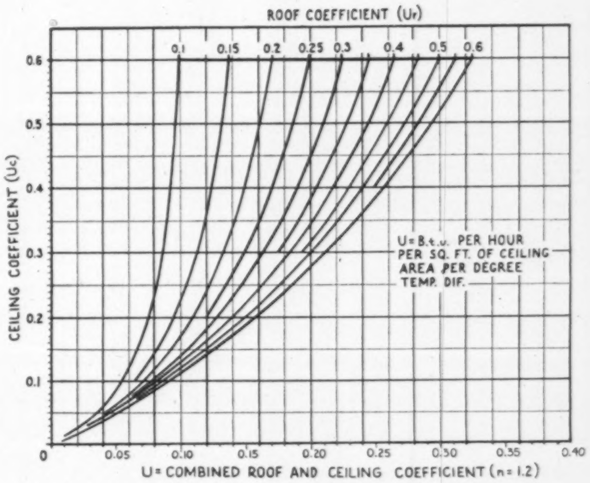

Fig. 2—Combined roof and ceiling coefficients for non-parallel roof and ceiling surfaces based on 1/3 pitch roof (See Note for Fig. 1.)

Equation (2) for a ⅓ pitch gable roof for various roof and ceiling coefficients, for a fixed attic wall coefficient of 0.36 and for inside and outside temperatures respectively of 70 F and 0 F. The attic is assumed to be without louvers, vents or other openings which would permit any air leakage, and the effect of sunshine is neglected.

Referring to Fig. 4, as would be expected, any decrease in the ceiling coefficient for the same roof coefficient, decreases the attic temperature, since less heat is available to warm the attic air due to the lower ceiling heat loss. Thus if the roof coefficient is 0.5, the attic temperature will be about 30 F when the ceiling coefficient is 0.5 (based on zero outside) and about +9 F when the ceiling coefficient is 0.1. On the other hand, a *decrease* in the roof coefficient for a constant ceiling coefficient *increases* the attic temperature, as would be expected. Take for example, a constant ceiling coefficient of 0.5: the attic temperature is about 30 F when the roof coefficient is 0.5 and about 50 F (+) when the roof coefficient is 0.1. In every day language, this means that with a well-insulated ceiling and a non-insulated roof, the attic will be cold whereas with the reverse situation, a well-insulated roof and a non-insulated ceiling, the attic will be warm. These facts are, of course, obvious to anyone familiar with the fundamentals of heat transfer.

Another interesting sidelight is that the effect of a decrease in the ceiling coefficient is partially offset by the decrease in attic temperature. For example, a reduction of the ceiling coefficient from 0.5 to 0.1 reduces the attic temperature from about 30 F to + 9 F for the conditions assumed in the example previously cited. Since the temperature difference through the ceiling is increased from 70-30, or 40 deg to 70-9, or 61 deg, the value of the lower ceiling coefficient is partially offset. This does not in any way discredit the value of the ceiling insulation but it does mean that the over-all effect of the insulation is not in proportion to the ratio of the reduction in the rate of heat flow through the ceiling. This fact can also be verified by means of Fig. 2 which shows, for example, that if the roof and ceiling coefficients are both 0.5, the combined coefficient will be 0.275 and if the ceiling coefficient only is reduced to 0.1, the combined coefficient will be about 0.085. This means, of course, that a reduction of the ceiling coefficient from 0.5 to 0.1 while the roof coefficient remains constant at 0.5, reduces the combined coefficient, not in the ratio of 5 to 1, but from 0.275 to 0.085, or in the ratio of 3.23 to 1.

Table 1—Values of *n* for Various Roof Pitches

Roof Pitch	*n*
¼	1.13
⅓	1.2
½	1.4
⅔	1.7
1	2.2

Fig. 5 shows the relationship between the attic temperature and the outside temperature for various combinations of roof and ceiling coefficients. For example, if the outside temperature is —10 F, the attic temperature will be about 0 F if the roof and ceiling coefficients are 0.5 and 0.1 respectively, whereas, if these coefficients are reversed the attic temperature will be about 45 F.

Typical coefficients of approximately the foregoing values are as follows: A roof of wood shingles on wood strips has a coefficient of 0.46 and a roof of asphalt shingles on wood sheathing has a coefficient of 0.56. In other words, a 0.50 roof is approximately an average of these two values. The 0.46 roof with 2 in. of corkboard applied to the underside of the roof rafters, has a coefficient of 0.10.

A ceiling construction having a value of approximately 0.50 is that

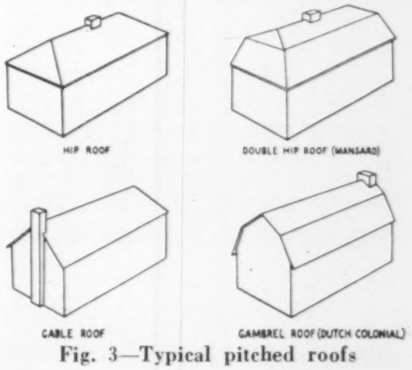

Fig. 3—Typical pitched roofs

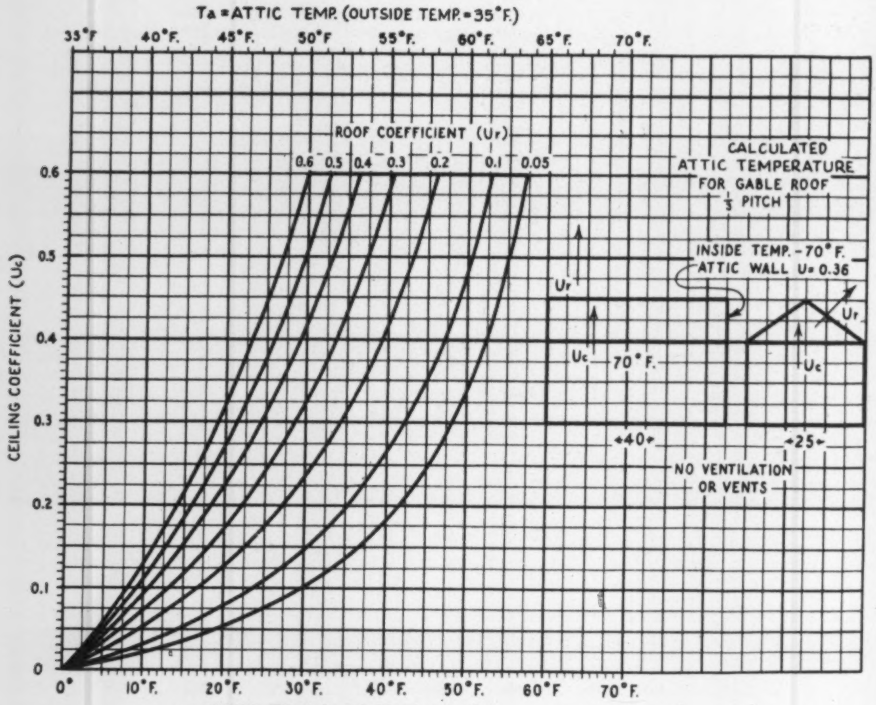

Fig. 4—Calculated attic temperatures for gable roofs

Note: These curves are based on the following conditions: Roof area, 1200 sq ft; Ceiling area, 1000 sq ft; Attic walls, 208 sq ft; Attic wall coefficient, 0.36; Inside room temperature, 70 F; Outside temperature, 0 F.

consisting of 3/8 in. plasterboard and 1/2 in. plaster with aluminum foil applied to the upper or back surface of the plasterboard, the coefficient being 0.53. A metal lath and plaster ceiling insulated with 2 in. of flexible insulation with an air space between has a coefficient of 0.10 (see ASHVE Guide 1942, Chapter 4, Table 8).

Air Leakage and Attic Ventilation

In the previous discussion, no mention has been made of the effect on temperatures in unheated attics of infiltration of air through loosely fitted or open windows, louvers, vents or other means of air leakage or ventilation. Obviously the effect of infiltration or ventilation is to reduce the attic temperature, the greater the amount of air change, the greater the reduction in the winter air temperature and consequently the greater the ceiling heat loss due to the increase in the temperature difference.

The exact effect of the air leakage or ventilation upon attic temperatures may be problematical in most cases because of the uncertainty of the volume of air entering the attic per hour but the general statement can be made that the attic temperature due to ventilation will range somewhere between the normal attic temperature without any interchange of air and the outside air temperature. If, however, an estimate can be made of the volume of air entering the attic, this can be translated into degrees temperature change by means of the following modification of Equation (2) which includes an air leakage factor:

$$t_{av} = \frac{A_cU_ct_i + t_o\,(A_rU_r + A_wU_w + A_gU_g + 0.018Q)}{A_rU_r + A_wU_w + A_gU_g + A_cU_c + 0.018Q} \quad \ldots (2a)$$

where

t_{av} = attic temperature with ventilation, degrees Fahrenheit

Q = volume of outside air entering attic per hour, cubic feet

If the attic temperature is calculated by means of Equation (2) which neglects ventilation or air leakage and then calculated by means of Equation (2a) which allows for ventilation, the difference ($t_a - t_{av}$) will be the number of degrees temperature drop in the attic due to the ventilation.

An example will illustrate the point. A gable roof has an area of 1200 sq ft and a coefficient of 0.5. The ceiling area is 1000 sq ft and the coefficient is 0.1, whereas the two end wall areas total 208 sq ft and the coefficient is 0.36. There are no glass areas in the gable ends. If the inside temperature is 70 F and the outside temperature is zero, what will be the attic temperature on a cloudy day if there is no air leakage into the attic? In this case,

$A_c = 1000$ $U_c = 0.1$ $t_i = 70$
$A_r = 1200$ $U_r = 0.5$ $t_o = 0$
$A_w = 208$ $U_w = 0.36$

Substituting these values in Equation (2):

$$t_a = \frac{1000 \times 0.1 \times 70 + 0\,(1200 \times 0.05 + 208 \times 0.36)}{1200 \times 0.5 + 208 \times 0.36 + 1000 \times 0.1} = 9.02 \text{ F}$$

Now assume the volume of air change due to air leakage or venting to be 1.5 cu ft per square foot of ceiling area per hour or $Q = 1500$ cu ft per hour. Substituting in Equation (2a), the recalculated temperature t_{av} will be 8.72. The difference between 9.02 and 8.72, or 0.3 deg represents the temperature drop due to the air leakage. This may seem to be a very small temperature change but this result can easily be

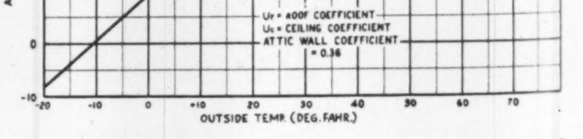

Fig. 5—Calculated attic temperatures for various roof (U_r) and ceiling (U_c) coefficients and outside temperatures

Note: These curves are based on the following conditions: Roof area, 1200 sq ft; Ceiling area, 1000 sq ft; Attic walls, 208 sq ft; Attic wall coefficient, 0.36; Inside room temperature, 70 F.

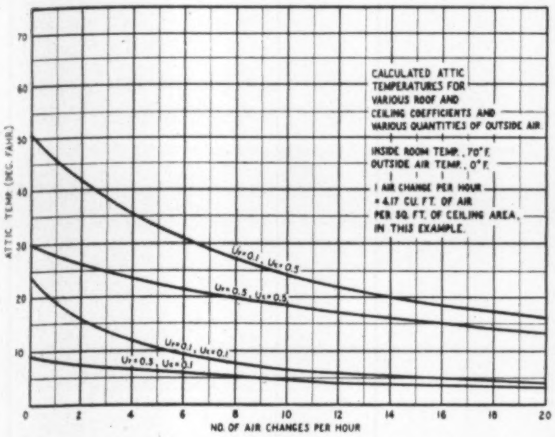

Fig. 6—Calculated attic temperatures[a] for various roof and ceiling coefficients and for various quantities of outside air expressed in air changes per hour[b]

[a] Data similar to that for Figs. 4 and 5, except as noted.
[b] One air change = 4.17 cu ft of air per square foot of ceiling area per hour in this example.

verified by balancing the heat flow through the ceiling with the heat flow through the roof and gable walls and the heat loss due to air leakage, on the basis that the heat lost through the ceiling must equal that lost through the gable walls, the roof and that due to air leakage. The heat transmitted through the ceiling will be 1000 × 0.1 × (70 — 8.72) or 6128 Btu per hr. The heat losses through the roof, gable walls and that due to air leakage are as follows:

Roof: 1200 × 0.5 × (8.72 — 0) = 5232
Gable Walls: 208 × 0.36 × (8.72 — 0) = 654
Air Leakage: 0.018 × 1500 × (8.72 — 0) = 235

Total 6121

The small discrepancy is due to the fact that the calculations were carried to only two decimal places. The reason for the comparatively minor effect of the ventilation on the attic temperature is due to the relatively small heat equivalent of the 1500 cu ft of air entering from the outside which amounts to only 235 Btu per hour or only 3.8 per cent of the total heat loss. If the volume of air entering per hour were 10 times as much as in this example, or 15,000 cu ft per hour (15 cu ft per sq ft of ceiling area), the attic temperature would be 6.7 F and the temperature drop due to the ventilation would be 9.02 — 6.7, or 2.32 deg, which is still a comparatively small amount. As the attic volume is 4167 cu ft in this case, this amounts to $\frac{15,000}{4,167}$ or 3.6 air changes per hour.

Attic Condensation and Venting

The original air change volume of 1.5 cu ft of outside air per hour per square foot of ceiling area is an amount sufficient to preclude attic condensation according to University of Minnesota tests. These calculations also check with data obtained at the University of Minnesota showing that the increased heat loss due to small air changes such as is required to preclude attic condensation is negligible in most cases, and can be neglected without serious error. Incidentally, the term *ventilation* as applied to such applications is a misnomer as this word generally implies a considerable interchange of air. For this reason, *venting* would probably be the more appropriate term.

Fig. 4 further illustrates the fact that the attic temperature is a function of the ceiling and roof coefficient. The proper combination of ceiling and roof materials will result in attic surface temperatures which will be above the dew-point at all times and will theoretically obviate the necessity for venting. Many engineers however regard the venting of insulated attics as good practice in all cases in cold climates.

Variation of Attic Temperature with Volume of Outside Air

Fig. 6 shows the relationship between the volume of outside air entering the attic per hour and the attic temperature, for inside and outside temperatures of 70 F and 0 F respectively and for various combinations of roof and ceiling coefficients. It should be noted that these curves are based on considerable quantities of outside air, expressed in terms of the number of air changes per hour. One air change provides from two to three times as much air as is generally required to preclude attic condensation.

Conclusions

Following are some of the conclusions relative to winter heat losses through unheated attics which may be drawn from this analysis:

1. The heat losses through hip and double hip roofs and unheated attics may be calculated by using the combined roof and ceiling coefficients, the ceiling area and the inside and outside temperature difference. The combined coefficient method of calculating ceiling heat losses can also be used with reasonable accuracy in many cases with gable roofs.
2. Another method to be used with roofs other than hip and double hip types is to calculate the attic temperature and to estimate the heat loss through the ceiling using the ceiling coefficient and area and the temperature difference through the ceilings.
3. Variations in the ceiling and roof coefficients vary the attic temperature. A decrease in the ceiling coefficient increases the attic temperature and vice versa, and a decrease in the roof coefficient increases the attic temperature and vice versa.
4. A decrease in the ceiling coefficient is partially offset by the increase in temperature difference through the ceiling which in effect means that the reduction in the ceiling heat loss is not in the ratio of the reduction in the ceiling coefficient but somewhat less, the exact ratio depending on conditions.
5. Attic air leakage and attic ventilation reduce attic temperatures, but small air changes such as are adequate to preclude attic condensation do not appreciably reduce the attic temperature and may, therefore, be neglected. To obtain an appreciable reduction in the attic temperature, several air changes per hour are required.
6. Attic condensation is a function of the attic temperature (since the condensation depends on the dew-point). Increasing the attic temperature, therefore, decreases the possibility of attic condensation and vice versa, but in any case attic condensation can be avoided by the proper use of vapor barriers and/or attic venting.

APPENDIX

Actual Attic Temperature Data

In order to check the accuracy of the calculated results, attic temperature readings, together with simultaneous inside and outside temperatures, were recorded during the 1941-1942 heating season, for a residence located in Kenilworth, Ill., a suburb of Chicago. These actual attic temperature readings were compared with the calculated attic temperatures for the prevailing inside and outside temperatures, the calculations being based on Equation (2). This formula reduces to the following in the case of this residence:

$$t_a = 0.21\, t_i + 0.79\, t_o$$

This residence has a gas-fired mechanical warm-air heating system which is thermostatically controlled. The ventilating louvers were closed so that the results are comparable with the calculated results for this condition only. The temperatures were read to the nearest degree.

No allowance was made in the calculated results for solar radiation and it was, therefore, possible in a measure, to evaluate the effect of this factor. Fig. A is a typical chart for two average winter days from 7 a.m., November 24 to 11 p.m., November 25, 1941. It will be noted that the actual and calculated attic temperatures check within 1 to 2 deg from about 11 p.m. until about 6 a.m. of this period. The outside temperature then rose rapidly but the attic temperature lagged behind somewhat so that there was a difference of several degrees from about 7 to 10 a.m., on November 25 between the actual attic temperature and the calculated attic temperature based on the inside temperature and the more rapidly rising outside temperature. The morning of this day (November 25) was bright and sunshiny, but the effect of the solar radiation did not become apparent until about 10:30 a.m., when the actual attic temperature curve crossed the calculated attic temperature curve.

As previously stated, no allowance was made in the calculated temperatures for solar radiation, so that the difference between the two temperatures after 10:30 a.m. was due largely to the effect of the solar radiation. The actual attic temperature reached the peak of 50 F at 2 p.m., about one hour after it had become cloudy. At this point the difference between the actual and calculated temperatures was 6.3 deg which difference probably represents the approximate effect of the solar radiation. The attic then began to cool off and did not reach the *normal* attic temperature (that is, the calculated temperature without allowance for solar radiation), until about 7 p.m., or nearly three hours after sunset. Thereafter, the two temperatures checked closely until the next day when solar radiation again began to influence the attic temperature.

One interesting result of these tests was the effect of solar radiation on the attic temperature on a cloudy day. Even with 0 per cent sunshine, there was a noticeable increase of 2 to 3 deg in the attic temperature during the daytime due to solar radiation. It should be pointed out, however, that as heating engineers are aware, the warming effect of solar radiation is not considered in estimating heat losses since the heating plant must be of sufficient capacity to maintain the desired inside temperature at night. Solar radiation does, however, lessen fuel consumption.

The calculated attic temperatures checked reasonably well during the night with the actual attic temperatures when the outside temperature was above 20 F, except when the outside temperature was changing rapidly, in which case the attic temperature would lag as would be expected. When the outside temperature was below 20 F, there was frequently a considerable discrepancy between the actual and calculated temperatures, the actual being considerably higher than the calculated temperature, this discrepancy increasing with decrease in outside temperature to 10 deg or more. However, it is believed that a sufficient number of actual attic temperature readings were taken to make certain that the calculated results could be assumed to be reasonably accurate for steady-state nighttime conditions and for outside temperatures of 20 F or higher. Further verification for outside temperatures below 20 F is necessary. The calculated results also checked closely with data obtained at the University of Minnesota and reported in Bulletin No. 18.[1]

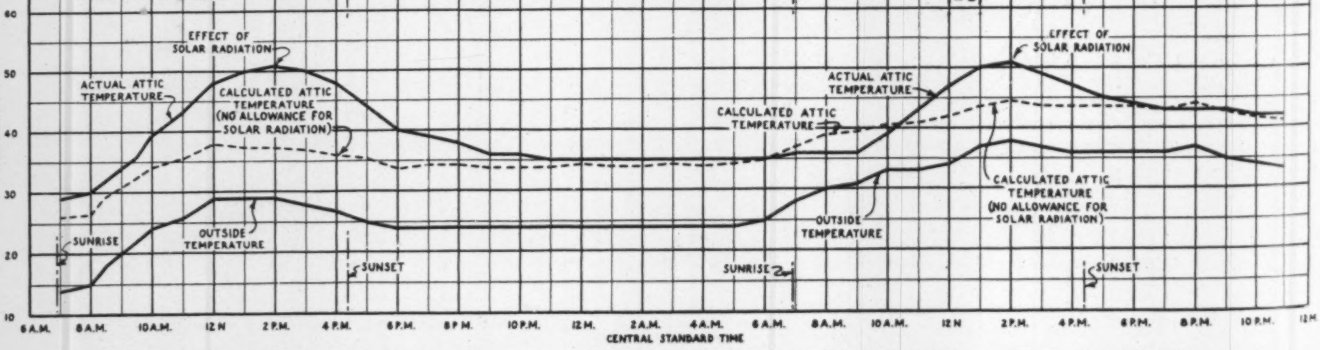

Fig. A—Typical chart for two average winter days from 7 a.m., November 24 to 11 p.m., November 25, 1941

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[1] Condensation of Moisture and Its Relation to Building Construction and Operation, by Frank B. Rowley, Axel B. Algren and Clarence E. Lund, (Bulletin No. 18 of the University of Minnesota Engineering Experiment Station, September 1941).

War On Fuel Waste

By B. M. Woods, Berkeley, Calif.
Chairman of ASHVE War Service Committee

With the extension of fuel oil rationing to the middle west in September the Fuel Conservation Campaign sponsored by the Society gained impetus, and the booklet prepared for the guidance of home owners has been widely quoted and 25,000 copies have already been distributed.

Conservation of all heating media (coal, coke, gas, oil, electricity and wood) is recognized as being important to war production today and every Society member has a duty to see that the *War on Fuel Waste* is carried on in his own community.

Figures issued for 48 states and the District of Columbia by the Bureau of Census give a comprehensive picture of the types of fuel used for heating dwellings throughout the United States. In dwellings having central heating plants, New York has 58.8 per cent burning coal and coke; 12.6 per cent using oil, and 1.9 per cent burning gas. The District of Columbia, Connecticut and Maryland report the highest percentage of oil burning installations, whereas Oregon, Maine and Vermont show the highest percentage of wood-burning equipment, and California, Montana, Wyoming and Kansas lead in gas heating installations.

Soon after the Society offered its research facilities to various Government Agencies, the Secretary of the Interior requested the ASHVE to examine and report on matters relating to conversion of oil-heating equipment to other fuels and the use of substitute materials for critical metals. This matter was referred to the War Service Committee for attention, and on June 18 it drafted the Committe's findings and recommendations which were supplemented on July 3 and presented to the Secretary of the Interior with the suggestion that immediate action be taken to announce a specific plan to the public, as there were only 90 days until the heating season commenced in some of the areas which were considered critical.

Between July 15 and October 15, various spokesmen for OPC, WPB, ODT, OPA and material issued by OWI carried alternate optimistic and pessimistic statements regarding oil shortage, transportation and pipe-line construction, release of grates for conversion, and fuel oil rationing, with the result that the public mind has been badly confused and seriously inadequate action has been taken. The absence of a clear and harmonious statement of policy from responsible Government sources has hindered progress. The complexity of the problem is made evident by the many statements issued by the agencies mentioned. The public is itself also at fault, since every optimistic statement has been overstressed.

On September 17, Price Administrator Leon Henderson announced that coupon rationing of fuel oil in 30 Eastern and Mid-Western States and the District of Columbia would begin October 15, "but will cover all oil used after October 1."

"The program will be initiated as soon as necessary forms and coupon books can be printed and distributed," he said,* warning bluntly that the prospect of uncomfortably cold homes this winter confronts residents of the rationed area who depend on oil for heating. The affected area, comprising more than a third of the country, contains almost three-fourths of the nation's population.

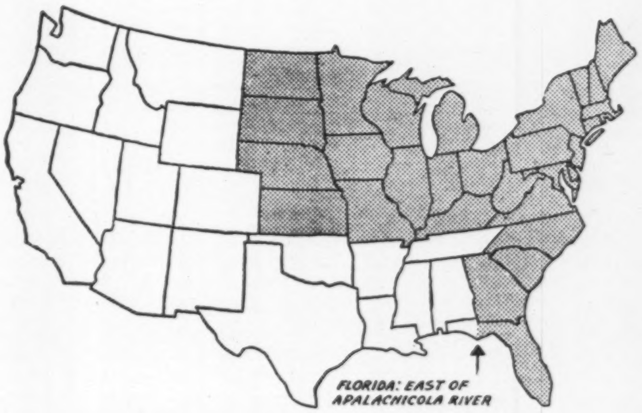
"The amount of oil to be allowed each heating unit under the plan, which is to cut consumption 25 *per cent* in the affected area, will be based on several factors, chief among them being floor space in the house and the amount of fuel used last year," he said.

On October 1, OPA announced two important changes in the fuel oil rationing plan, namely: The overall or average cut would be 33⅓ *per cent* from normal consumption instead of 25 per cent, and all buildings, other than residences, are to be cut the uniform 33⅓ per cent, instead of basing supplies on square foot of standing radiation. In making the deeper cut the Office of Price Administration announced that it was governed by the oil supply estimates of the WPB.

In describing the rationing objectives, Joel Dean, Director of Fuel Division, OPA, said that the program centered around the accomplishment of the following:

"(1) curtailment of fuel oil to the extent demanded by the petroleum transportation shortage; (2) equitable distribution of available supplies; (3) stimulation to convert to coal or to conserve by improving burner efficiency and by reducing structural heat loss; (4) consideration of health factors, age factors, and special heating requirements.

"The rationing covers all fuel oils (both distillate and residual, grades No. 1 through No. 6, including



Shading denotes fuel oil rationed states

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*Announcement was made October 15 by Mr. Henderson that distribution of ration coupons would commence October 22.

diesel fuel and kerosene) and includes all used (home heating, industrial, commercial, military and miscellaneous uses).

"Three factors determine the basic home fuel oil ration: (1) a standard *heat-loss formula* based on the square foot dwelling area to be heated; (2) the *previous heating season's consumption* reduced by the required percentage; (3) the *number of occupants* in each house. The ration will be adjusted to the normal temperature in the area by the use of a degree-day temperature table for each zone.

"To provide for disparity of climatic conditions throughout the rationed area, there will be four thermal zones.

"The heating season for each zone will be divided into five periods of approximately equal degree-days, which means that the periods are similar in respect to fuel needs.

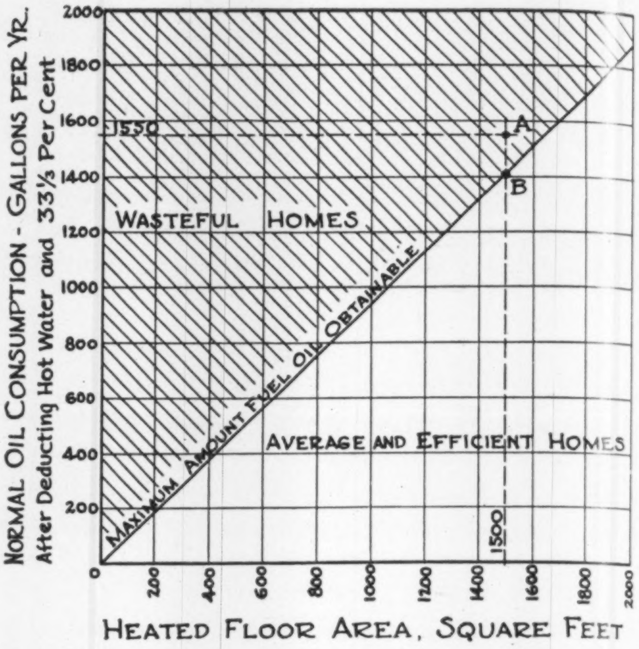
"With the heating season divided into five equal degree-day periods, tentative coupon expiration dates, starting from October 1, are:

Period	Zone			
	A	B	C	D
1	Nov. 29	Dec. 2	Dec. 3	Dec. 5
2	Jan. 6	Jan. 5	Jan. 4	Jan. 2
3	Feb. 8	Feb. 6	Feb. 2	Feb. 5
4	Mar. 17	Mar. 12	Mar. 6	Mar. 6
5	Sept. 30	Sept. 30	Sept. 30	Sept. 30

"A basic ration will be determined for the entire heating season. This ration will be determined in accordance with the previously mentioned rationing principles, within the limits of practical administration.

"The basic ration for heating private homes will be determined by two criteria: (a) A fuel-requirements formula which is based on the floor area of the home, the temperature of the locality, and a constant which reflects heat loss from a standard home. This formula will set the ration range as described later. (b) A percentage of past consumption adjusted to normal weather conditions will determine the ration if the applicant falls within the range. Users who fall below the lower limits of the formula range are unusually efficient. They will be cut less deeply but they will under no circumstances receive more than 85 per cent of last year's consumption adjusted to normal."

The formula which expresses fuel requirements of a typical free-standing house of average construction and average burner efficiency modified by the normal temperature of the particular area is $F = \dfrac{AD}{K}$, where F is the fuel requirements, A floor area, D is the normal degree-days at that locality, and K is a constant determined through empirical tests to be the standard of thermal efficiency. This formula gives the total number of gallons of fuel oil required for the structure. OPA has assumed that K (the denominator in this formula) can vary over a range from 5,600 to 7,300 without proving significant deviations from reasonable or standard efficiency. When the denominator for a given dwelling is under 5,600, however, the building is probably operating below standard efficiency and there may be a number of heat loss factors which can be corrected by the owner. At this point it becomes necessary to penalize the wasteful user.



For New York and Vicinity only—slope of diagonal varies with different cities

In checking several hundred houses OPA found that 25 per cent of them have *average* thermal efficiency so they adopted as part of the formula a *range* which covers this 25 per cent. Houses with thermal efficiency falling within this range (measured in gallons per square foot of floor space) will get a straight 33⅓ per cent cut from last year's consumption, adjusted to normal.

Better houses, with higher thermal efficiency, will not be cut as much. They will get the bottom limit of the *range* or a 15 per cent cut from normal, whichever is lower.

Commercial and industrial users of oil for heating water will also get a flat one-third cut, except that restaurants may get supplemental rations for this purpose if it is essential for health and sanitation.

The reduction in oil allowed to apartment houses, commercial, institutional and industrial establishments for space heating will be a straight 33⅓ per cent from last season, adjusted to normal.

The formula is designed to bring the room temperature of the standard home to a uniform temperature throughout the rationed area. The degree drop in temperature will depend upon the extent of the fuel oil shortage. Presumably, the standard temperature will not be less than 65 deg, which has been determined as the minimum health standard by OPA.

Fuel oil for heating water for domestic use will be curtailed in the same proportion as for space heating. The water heating ration will be added to the basic ration and will be based on 15 gal plus 5 gal per person per month.

Local rationing boards are being furnished with a book of instructions with various charts, so that they can make quick calculations to determine the fuel oil ration.

The Society was requested by Mr. Dean to supply the names of ASHVE members in the rationed areas and 1600 names have been submitted so that ration boards may call them for consultation.

The fuel rationing plan has been presented to the public in piece-meal fashion and comments on it in the trade and technical press have been far from flattering. In studying the plan it is obvious that the result is a compromise between the advocates of a *flat percentage cut* to all users and a *variable cut* which would reward the efficient plant. By the use of a mid-point denominator in the rationing formula for all new installations they generally will obtain a better ration than the existing plant that has been operating efficiently in the past. The inclusion of a range in applying the formula discriminates against the more efficient plant that has been operated economically in the past. This is regrettable, but probably unavoidable. The method indicated for determining the fuel ration for multiple dwellings gives no indication how supplemental rations can be provided for infants, invalids and aged persons, without definitely favoring the apartment dweller, as compared with the single house owner. It is not clear how oil consumption for cooking and heating can be separated, although it has been indicated that consumption of kerosene oil is of substantial volume, but little has been said about its control or rationing where it is used for heating purposes, so it is important for the public to realize that kerosene will be rationed.

The conclusion that a 65 deg temperature will prevail under the application of the formula is not justified. The formula is not based on a 65 deg standard temperature and may give very different results. The failure to determine and to publicize a specific plan during the summer months, so that individual home owners might determine their house efficiency rating is really unfortunate. It is likely to have serious consequences in the rationed areas.

Many requests have come to the Society from home owners, members of rationing boards and others, for interpretation of the rationing plan as announced up to October 15, and a simple chart to guide home owners has been prepared so that people could readily determine whether they were in the *rationing range*. Its use is explained in the paragraphs which follow.

A resident of the New York area will get a maximum amount of fuel oil under the announced rationing plan of 0.94 gal per square foot of floor area. This will be comparable to at least a cut of 33⅓ per cent and possibly as much as 40 or 50 per cent depending on how *wasteful* a house he has. Additional allowances will be provided for heating hot water, for children under 4 years of age, and for elderly persons.

For those having an *average* or *efficient* house the maximum cut for space heating will not be more than 33⅓ per cent and in some cases the cut will be as little as 15 per cent of last year's fuel consumption adjusted to normal winter weather conditions.

As last year was warmer than normal, in determining whether the cut will be greater than 33⅓ per cent, it is necessary to add 10 per cent to last year's oil consumption. Deduct from this the yearly hot water allowance which is computed by taking 15 gal per month plus 5 gal monthly for each occupant. Then subtract an additional 33⅓ per cent and if this result is *greater* than 0.94 times the number of square feet floor area, the latter is the rationed amount; but if *less*, then a more detailed procedure already outlined by the OPA in the press needs to be followed.

Suppose 2500 gal were used last season; there are four in the family; oil is also used for heating hot water 12 months in the year, and there are 1500 sq ft of floor area to be heated. Adding 10 per cent to last year's consumption gives 2500 + 250 = 2750 gal as normal. The hot water allowance is obtained by taking 15 plus 4 people times 5 = 35, multiplied by 12 months = 420 gal. Subtract this from the normal 2750 — 420 = 2330 gal. Reducing this by 33⅓ per cent gives a basic cut to 1550 gal for space heating. Enter the chart on the left hand scale with this figure of 1550 and draw a horizontal line. Where this line intersects a vertical line drawn for the proper floor area, point *A* results. Point *A* falls in the shaded portion of the chart above the diagonal line, so the rationed amount of fuel is point *B*, or 0.94 × 1500 = 1410 gal, which is equivalent to a cut of 39.5 per cent from normal. In this case it will probably be advisable for the person to take some immediate conservation measures or convert to coal, or otherwise the home will be uncomfortable.

It is good policy to do as much as you can as soon as you can, thereby doing your bit to *War on Fuel Waste* this winter and making a real contribution to increased war production.

Nominees for Committee on Research

(Three-Year Term)

H. J. Rose
Pittsburgh, Pa.

L. E. Seeley
New Haven, Conn.

L. P. Saunders
Lockport, N. Y.

A. E. Stacey, Jr.
Washington, D. C.

C. Tasker
Toronto, Ont., Canada

The Characteristics of Double Pitot Tubes

By F. R. Ingram,* E. Diez-Canseco,** and L. Silverman,† Boston, Mass.

SUMMARY—A study of three types of double chamber pitot tubes was made under different flow conditions. The use of double pitot tubes under conditions other than calibration may produce appreciable error. In pipes the coefficients were found to vary with the amount of turbulence present and to decrease with increasing air velocity. Outside of pipes under suction the coefficient was found to increase with increasing airflow. The magnitude of the coefficient depends considerably on the shape of the tube facing the airstream.

THE CYLINDRICAL or point type of pitot has been used for measurement of air flow inside and outside of openings (1, 2, 3).[1] This tube has the advantage of being able to measure velocities at points whereas the ordinary pitot tube static and total pressures are not measured at identical locations. To increase the sensitivity of this type of tube a double chamber tube is recommended. This tube has two independent chambers each of which has an impact hole one opposite the other. Hence in pipes this tube gives a greater velocity head reading because the flow conditions create a negative pressure on the downstream side of the tube in addition to the impact pressure created on the forward tube. This increase in reading thus makes the tube more sensitive and applicable for lower velocity measurements. A very small cylindrical type (⅛ in. diam.), a modification of the Stauscheibe tube (5), was used for air flow measurements before openings under suction (1, 2). When calibrated in a pipe it showed coefficients ranging from 2.00 to 1.75. The value of the coefficient[2] was stated to be constant at velocities greater than 950 fpm and the pipe calibration was used for outside flow measurements. Measurements made in this laboratory showed that the coefficients varied appreciably with flow conditions and indicated further information would be of value.

This investigation was undertaken to study the effects of flow conditions, size, and shape of double tubes on their coefficients and to determine whether the coefficients were constant enough for practical use under various conditions of air flow in pipes and outside of openings under suction.

Description of Tubes

The tubes used to measure the velocities under the various flow conditions are shown in Fig. 1. They were constructed of brass to the dimensions indicated. The double round tubes were each separated into two chambers by means of a thin brass strip extending the full length of the tube. The double square and the double streamline tubes were made from solid stock and 1/16 in. brass tubing. Diametrically opposite holes, each 0.02 in., were drilled in the mid section of each double tube at the location shown in Fig. 1. A single round ⅛ in. pitot tube was used in place of the double ⅛ in. tube used in previous studies because of construction difficulties. Comparable results were obtained by adding the reading obtained with the opening facing the flow to that obtained by rotating 180 deg when the opening was facing downstream.

Experimental Equipment

Various flow conditions were used in the study in order to determine whether the tube characteristics were affected by calibration conditions.

Condition 1—Rectangular duct: Readings were made at the centerline each way in a 9½ in. x 16 in. rectangular duct, 6 ft from the inlet. A large bell-mouthed flange was provided on the inlet in order to give uniform flow conditions. An 8 in. pipe section with a venturi (5 in. throat) connected the rectangular duct to a planing mill exhaust fan whose discharge was conducted to the outside. Flow regulation was obtained by means of a damper at the fan inlet.

Condition 2—Round tunnel: This system was primarily the one described for *Condition 1*. However, the rectangular duct portion of the system was replaced by a 6½ in. diameter round pipe. A 15 in. diameter flange was used at the inlet and a straightening section (30 pieces of 1 in. diameter x 24 in. brass pipe) was placed in the pipe 2 diameters from the inlet. Readings were taken after the straightening section 8.7 diameters from the inlet. A piezometer ring with five taps was used to obtain static pressures.

Condition 3—¾ in. pipe: In order to determine the pitot characteristics under streamline flow (Reynold's No. <1300), a ¾ in. inside diameter brass pipe was fitted with static taps. The measuring station was located 31 diameters from the mouth of the pipe. Suction was provided by a small suction pump with a bleeder valve. Quantities were obtained by a thin plate orifice inserted in the duct below the pitot station.

Condition 4—Pipe entrance: Measurements under this flow condition were made one diameter from the inlet of a 4 in. pipe provided with a 16-in. diameter flange. Static taps, 90 deg apart, were connected as a piezometer ring and used to measure the static pressure. A standard 3 in. thin plate orifice inserted in the system indicated the air volume. A blower of 240 cfm capacity was used to provide the required suction. Air volumes being controlled by a damper on the fan discharge outlet.

Condition 5—Outside pipe: This system was the same as that used for *Condition 2*. Measurements were

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**Graduate Student, Department of Industrial Hygiene, Harvard School of Public Health.

†Associate, Department of Industrial Hygiene, Harvard School of Public Health.

[1] Numerals in parentheses refer to Bibliography.

[2] For water flow it is stated (4) that the coefficient is never twice that obtained with a pitot-static tube but only 35 to 40 per cent greater depending upon the design of the tube, the fluid velocity, and position of tip in the channel.

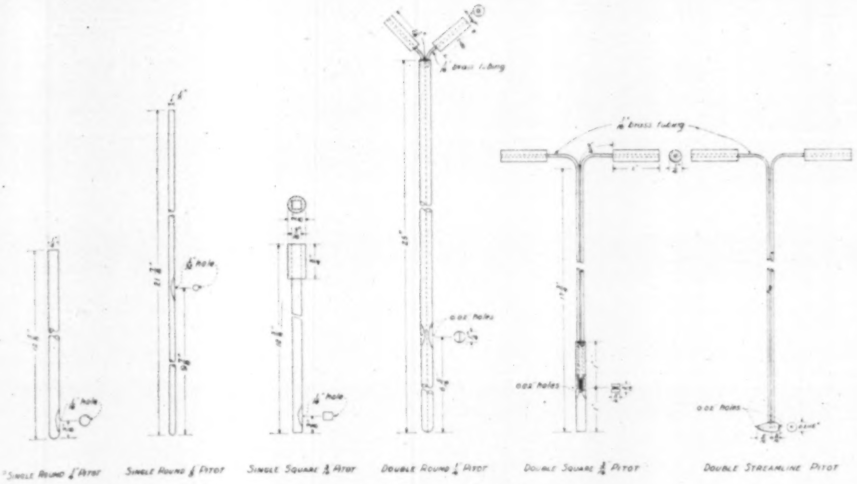

Fig. 1—Types of tubes used in investigation

made at the centerline of the pipe and 4 in. in front of the flanged opening. Velocities outside of the pipe were checked with the heated thermometer anemometer (6), and a standard Kata (the manufacturers calibration was used).

For high velocities, a vertical water manometer was used for measurements. For lower velocities an inclined manometer (scale 1 to 10) was used and for very low readings, the Wahlen gage (7) was employed. Several readings were taken on the latter and averaged.

Determination of Coefficient

The coefficient[3] which must be applied to the observed manometer reading in order to obtain the true velocity head for the formula $V = K\sqrt{H_v}$ was determined for each type of pitot tube as follows:

1. The value of h_v was determined. This is the value recorded by the front chamber of the pitot tube and is the difference between the total pressure (from the front chamber to one leg of the manometer) and the static pressure (from the static tap to the other leg of the manometer).
2. The value for h_v' was determined. This is the value recorded as the difference between the total pressures on the front and rear chamber of the pitot tube. It is measured by connecting the front chamber to one leg of the manometer and the rear chamber to the other leg of the manometer and noting the difference. The static pressure being the same in each case, the manometer reading represents the sum of the velocity head on the front chamber and the negative pressure produced in the region of the rear chamber opening by the air flow.
3. The coefficient desired, therefore, is the ratio of this summation of pressures to the front chamber velocity pressure.

Flow Inside Pipes

Table 1 shows a comparison of the modified NPL standard pitot tube (8) with the round ¼ in. cylindrical impact tube. Velocities obtained with the round ¼ in. impact varied from 4 per cent below to 3.25 per cent above those obtained with the modified NPL standard pitot. The average variation was less than 2.25 per cent. These data indicate that a single round ¼ in. impact tube may be used for flow in pipes instead of the standard pitot with no appreciable error resulting from its use.

In Fig. 2 a graphical comparison is made of the single ¼ in. and ⅛ in. round and the $\frac{3}{16}$ in. square tubes under varying flow conditions. It is evident from a study of this graph that any of the tubes studied may be used under any of the pipe flow conditions without affecting the coefficient which is unity.

Table 1—Comparison of Modified NPL Standard Pitot Tube (8) with Single ¼ in. Round Cylindrical Tube in 6½ in. Pipe Calibration (9)

Modified NPL Standard FPM	Single ¼ in. Round FPM	Per Cent Deviation	Average Deviation
1438	1400	—2.65	....
1360	1305	—4.04	....
1149	1188	+3.22	....
1127	1109	—1.60	....
1060	1040	—0.88	....
861	869	+0.93	....
711	734	+3.24	....
681	670	—1.61	....
469	460	—1.92	2.23

In Table 2 are shown the values of coefficients of the various double tubes under the several flow conditions.

For the rectangular tunnel condition, the ¼ in. round double pitot gave a higher coefficient (1.70) than either the $\frac{3}{16}$ in. square (1.34) or the streamline (1.05) double pitot tubes, but the standard deviation was also higher (0.274 compared with 0.120 and 0.107 respectively) indicating that the reliability of the mean coefficient was much less than that for either the double square or the streamline tube. The same relative comparison may be made for the round tunnel. The coefficient found for the ⅛ in. round tube, however, is much higher than for the others being 1.92 (std dev. = 0.098) as compared with the ¼ in. round pitot which has the next highest coefficient of 1.63 (std. dev. 0.142). The results obtained with the ⅛ in. round pitot in the 6½ in. round tunnel compare very favorably with those for the same tube obtained by Dalla Valle (1, 2) in an 8 in. diameter duct (mean coef. = 1.84, std. dev. = 0.106).

In the ¾ in. pipe, calculations showed that with this small pipe laminar conditions existed at velocities less than 214 fpm; however the centerline velocity was not twice the average because the ratio $\left(\frac{x}{rR}\right)$ according to Boussinesq (10) was less than 0.26. This did not interfere with the authors' studies since coefficients were determined by actual measurements. The highest coefficient (2.03) was obtained with the $\frac{3}{16}$ in. double square pitot, with the next highest coefficient (1.83) being obtained with the ¼ in. round double pitot. The standard deviation for the square tube (0.163) was less than that for the round tube (0.181), indicating a greater reliability of the data obtained with the square tube under this flow condition. The streamline tube coefficient was 1.01 or practically unity. Its only advantage as compared to the ordinary standard pitot is the ease with which it may be inserted into a duct. This might be desirable for field measurements.

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[3] This coefficient is denoted by C. If the observed manometer reading is denoted by h_v then the velocity head H_v is equal to $\frac{h_v}{C}$ or $C = \frac{h_v}{H_v}$.

At the mouth of a flanged opening under suction a different condition of turbulence exists and the mean coefficient for the ¼ in. round tube is only 1.45 (std. dev. 0.134) compared to the $\frac{3}{16}$ in. square whose coefficient is 1.44 (std. dev. 0.105). The coefficient for the streamline tube under this flow condition is greater (1.16) but the reliability of the results obtained is decreased because of the turbulent conditions.

Table 3 shows a comparison of the coefficients covering all flow conditions for each type of double tube and for the ⅛ in. single round tube used as a double tube. The results are indicated in Figs. 3 and 4.

It can be seen in Fig. 4 that the values of the coefficient obtained at low flows in this work were lower than those obtained in (1). This deviation is due to difference of calibration conditions. The values given here were obtained in a 6½ in. pipe with flanged entrance and straighteners, while those in (1) were obtained in an 8 in. duct with a standard ASHVE tube 8 diameters down stream. No further specifications were given.

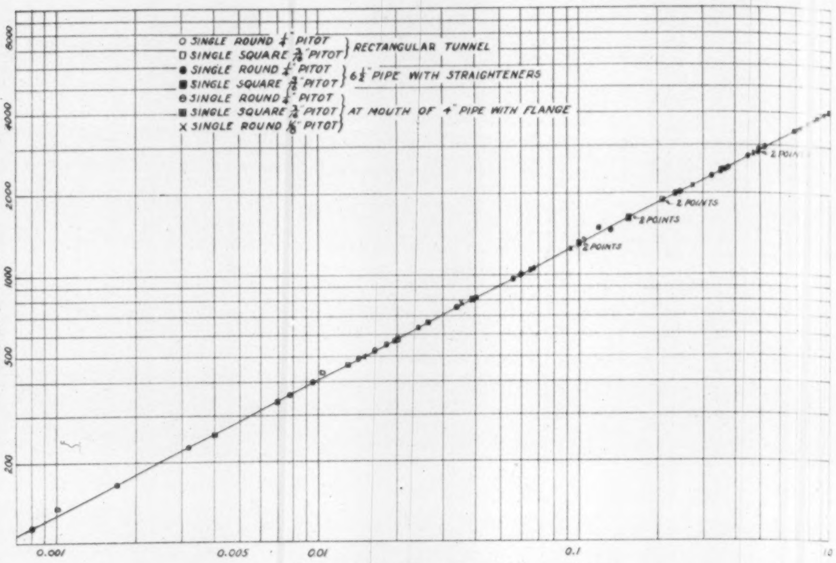

Fig. 2—Comparison of single round and square pitot tubes under varying conditions of flow

Location of Pitot in Pipe Traverse

In order to determine what effect the location of the pitot opening would have with relation to its posi-

Table 2—Values of Double Tube Coefficients under Various Flow Conditions

Double-Pitot Tubes	Condition Under Which Calibration Was Made											
	Rectangular tunnel			Round Tunnel			¾ In. Pipe			Pipe Entrance		
	No.	Mean	σ^a	No.	Mean	σ	No.	Mean	σ	No.	Mean	σ
¼ in. Round	16	1.70	0.274	18	1.63	0.142	6	1.83	0.181	14	1.45	0.134
$\frac{3}{16}$ in. Square	18	1.34	0.120	17	1.40	0.091	8	2.03	0.163	13	1.44	0.105
Streamline	19	1.05	0.107	15	1.02	0.049	4	1.01	0.039	6	1.16	0.106
⅛ in. Round	....	....	....	8	1.92	0.098	....	....	....	....	....	....

From (1). ⅛ in. double-pitot tube in 8 in. diameter round duct calibrated against standard pitot: No. Readings, 13; Mean Coef. = 1.84; σ = 0.106
$^a\sigma$ = Standard deviation.

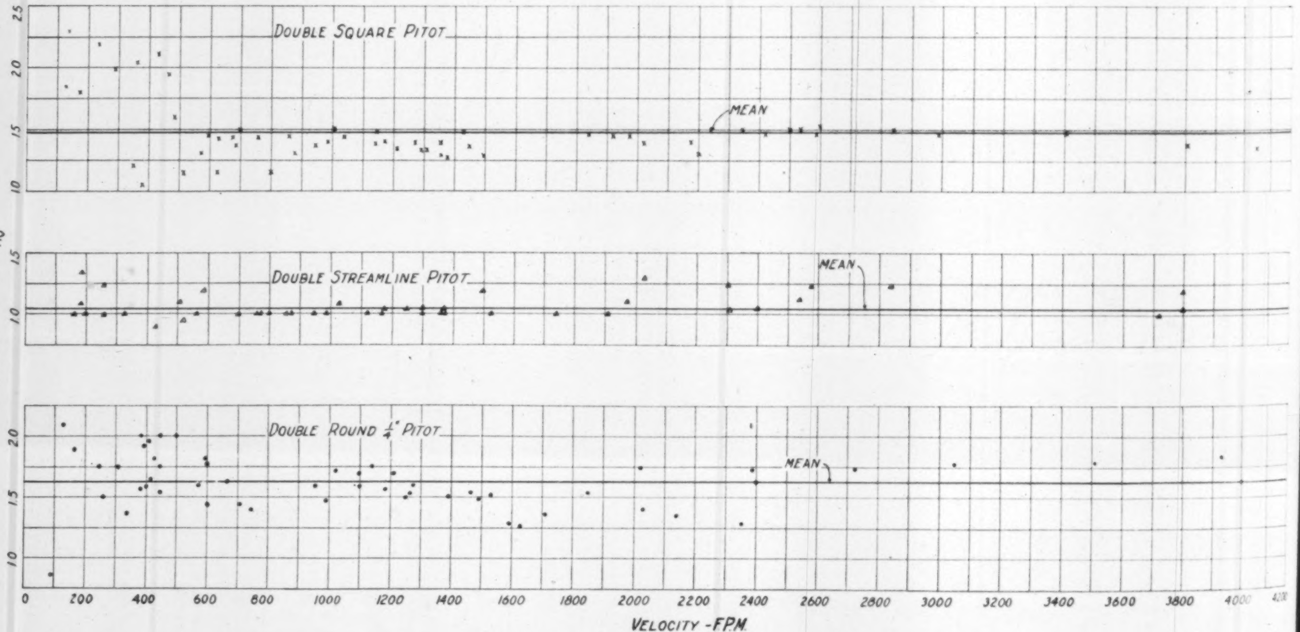

Fig. 3—Comparison of double-pitot tubes under varying flow conditions

Table 3—Over-all Coefficients Covering All Flow Conditions

TUBE	NO. READINGS	MEAN COEFFICIENT	σ
¼ in. Double Round	54	1.63	0.202
⅛ in. Double Square	56	1.48	0.255
Double Streamline	44	1.05	0.096
⅛ in. Round[a]	8	1.92	0.098
⅛ in. Double Round (from (1))	13	1.84	0.106

[a] Coefficient obtained by rotation as described above.

Table 4—Effect of Location in Pipe Traverse Double ¼ in. Pitot in 6½ in. Round Pipe with Straightening Section (9)

LOCATION OF PIPE	ACTUAL VELOCITY FPM	PITOT COEFFICIENT	AVERAGE
Centerline	1508	1.49	
	1304	1.53	
	1300	1.50	
	962	1.52	
	860	1.51	
	464	1.54	
			1.51
1 in. up from centerline	1440	1.60	
	900	1.71	
	392	1.72	
			1.66
2 in. up from centerline	1264	1.66	
	812	1.66	
	372	1.52	
			1.61
3 in. up from centerline	1000	2.34	
	702	2.17	
	275	2.08	
			2.20

tion along the cross sectional center line, studies were made with the double ¼ in. round pitot in the 6½ in. pipe provided with a straightening section. Readings were made for several air flows at the center line and at 1, 2 and 3 in. from the center line. The results of these studies are shown in Table 4.

The center line coefficient is slightly lower than that obtained at either the 1 or 2 in. distance, but the coefficient obtained ¼ in. from the pipe wall was relatively higher. This is due to flow conditions being more uniform adjacent to the wall of a pipe. Thus, it may be seen that little change in the coefficient is obtained if the pitot tube is immersed in the air stream to a distance less than ⅓ of the pipe diameter from the center. As a traversing instrument it would not be satisfactory to use, however, because of the variability in coefficient.

Rotational Characteristics of Double Pitot Tubes

The use of this type of tube as a direction meter was previously recommended by (1, 2). In order to determine the rotational characteristics of these tubes they were placed in the 6½ in. duct with a straightening section. A calibrated sector was used for angle measurements.

The rotational characteristics of the ¼ in. round, square, and streamline double pitot tubes are shown graphically in Figs. 5, 6 and 7. The per cent of velocity (when the openings are parallel to the direction of flow) is plotted against the angle through which the tube was rotated. Measurements were taken from 0 deg to 360 deg for each tube, but because of the symmetry of the round and square tubes the values were averaged and a single average curve from 0 deg to 180 deg is shown. This is not the case with the streamline tube and curves are shown for both conditions 0 deg to 180 deg and 180 to 360 deg (0 deg). It is noted that zero pressure readings occur only once for each type of tube during its rotation from 0 deg to 180 deg and that this zero reading does not occur at 90 deg as one might expect. Studies made by others (1, 2) on the ⅛ in. round double pitot tube show a zero pressure reading at three points (63 deg, 90 deg, 117 deg) during rotation from 0 deg to 180 deg. Our study indicates that the maximum reading of the tube occurs over a range of ± 15 deg and that the instruments are not particularly sensitive to direction in this range.

Flow Outside of Pipes

Values of coefficients obtained in the air stream outside of a pipe under suction are shown in Table 5. In column 2 the actual velocity as measured by a heated thermometer (6) is given and column 3 shows the

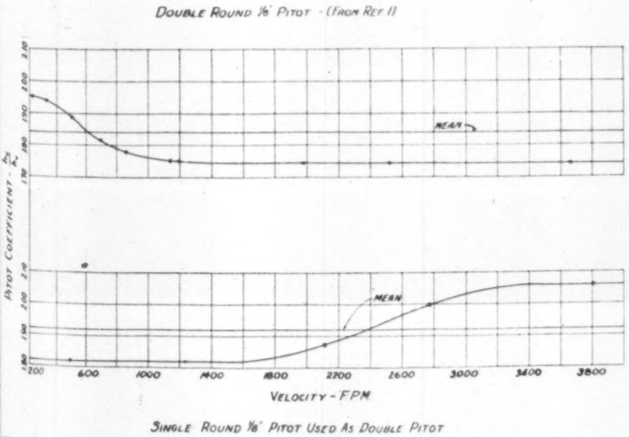

Fig. 4—Comparison of ⅛ in. round tube coefficients with previous data

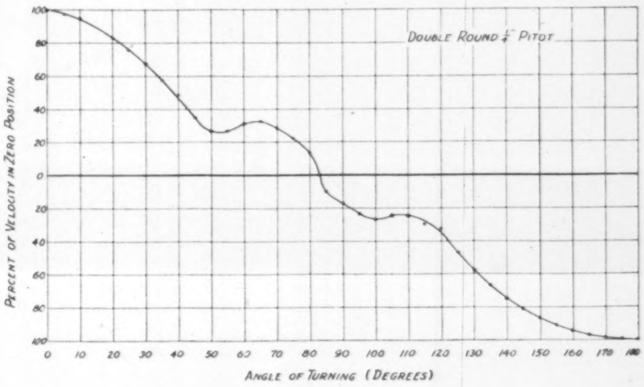

Fig. 5—Rotation characteristics of double ¼ in. round

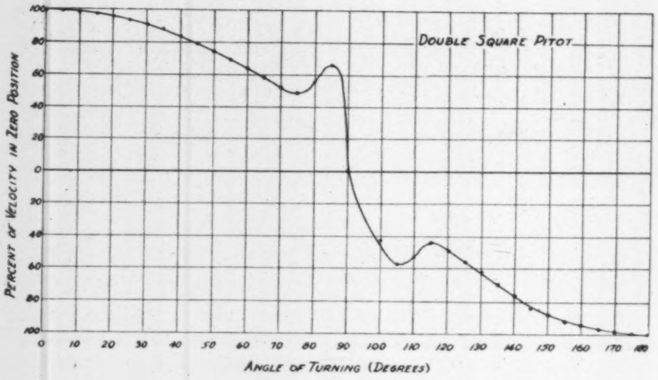

Fig. 6—Rotation characteristics of double $\frac{3}{16}$ in. square

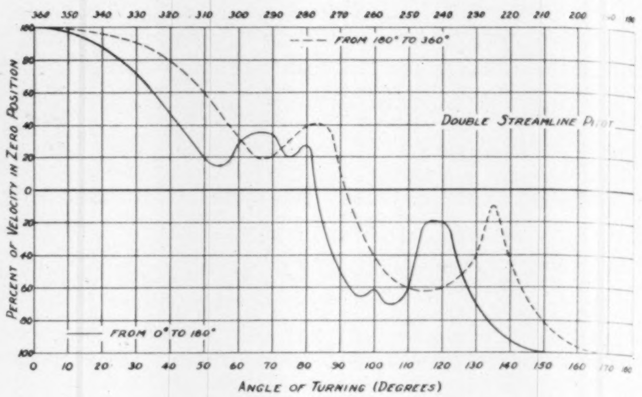

Fig. 7—Rotation characteristics of streamline shape

corresponding velocity head. These velocities were also checked by means of a standard Kata thermometer, as shown in Fig. 8. Column 4 indicates the reading obtained from the front chamber. Since this is an opening under suction the total pressure is zero; hence the values in this column can only be due to turbulence (both positive and negative values were obtained but for practical purposes they are negligible). Photographs of flow in Prandtl-Tietjens (10) indicate that there is a splitting of the fluid stream as it approaches any shape immersed under laminar flow conditions. Hence as the air approaches the tube in front of an opening it tends to create a region of turbulence in front of the impact hole. A check by means of a standard ASHVE type of pitot gave zero readings on impact pressure at this point; hence the values shown must be attributed to impact created by turbulence. Column 6 represents the readings obtained from the double tube connections. The values in column 7 show that there is a trend toward increased coefficient at increased velocity. The values in column 5 show that the static reading obtained by the rear tube may be more than twice the actual velocity head. This condition is created by the viscosity of the air moving close to the hole and the negative pressure zone in the wake of the hole. When a single tube was rotated 90 deg so the hole was perpendicular to flow direction high static readings were also obtained. These conditions are similar to the variations in air pressure around an airship hull as described in Ower (8). Even a streamlined object may show a small amount of turbulence in its wake. This is verified by the negative pressures recorded by the streamlined tube.

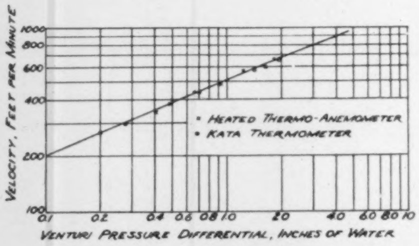

Fig. 8—Measuring station calibration 4 in. in front of flanged opening. Each Kata point is average of several measurements

It is significant that there is no uniformity or constancy for coefficients of a given tube, the coefficient increases as the velocity increases, being 2.0 or greater for velocities above 500 fpm, whereas in a pipe the opposite trend was noted. A comparison of the ¼ in. single round tube used as a double tube with the ¼ in. double round tube shows higher coefficients. This is explained by the fact that the single round tube has a larger hole ($\frac{1}{16}$ in.) than the double round tube (0.02 in.) thus creating different static conditions.

Conclusions and Summary

A simple impact tube used for airflow measurements in pipes of cylindrical or square shape gives comparable results with a standard type of pitot tube.

The use of double pitot tubes under conditions other than calibration may produce appreciable error.

Table 5—Values of Coefficients in Air Stream Outside of Pipe under Suction

Type of Pitot Tube	Velocity Measured by Heated Instrument fpm	h_{va} Corresponding to Heated Instrument In. W.G.	h_{vf} Recorded by Front Chamber of Pitot Tube In. W.G.	h_{vr} Recorded by Rear Chamber of Pitot Tube In. W.G.	h_v Recorded by Both Chambers of Pitot Tube In. W.G.	Double Tube Coef. Col. 6 / Col. 3	Rear Chamber Coef. Col. 5 / Col. 3
¼ in. Single Round......	300	0.0055	0.0013	0.0090	0.0103	1.87	1.64
	355	0.0078	0.0031	0.0134	0.0165	2.12	1.72
	418	0.0107	0.0034	0.0217	0.0251	2.35	2.03
⅛ in. Single Round......	288	0.00508	0.0014	0.0077	0.0091	1.79	1.42
	381	0.00890	0.0017	0.0141	0.0158	1.78	1.58
	438	0.01180	0.0021	0.0186	0.0207	1.75	1.58
	480	0.01410	0.0024	0.0244	0.0268	1.90	1.73
¼ in. Double Round......	218	0.00291	0.0007	0.0039	0.0046	1.58	1.34
	288	0.00509	0.0015	0.0070	0.0085	1.70	1.38
	340	0.00708	0.0022	0.0093	0.0115	1.63	1.31
	388	0.00921	0.0027	0.0135	0.0162	1.76	1.47
	525	0.01690	0.0020	0.3580	0.3600	2.13	2.12
	500[a]	0.0156	0	0.040	0.040	2.64	2.64
	585[a]	0.0214	0	0.051	0.056	2.50	2.75
	670[a]	0.0281	0	0.067	0.066	2.43	2.47
$\frac{3}{16}$ in. Double Square......	415	0.0106	0.0045	0.0143	0.0188	1.75	1.35
	490	0.0146	0.0011	0.0239	0.0250	1.71	1.64
	520	0.0166	0.0009	0.0370	0.0379	2.28	2.23
Double Streamline..	328	0.0066	0.0013	0.0061	0.0074	1.12	0.92
	438	0.0118	0.0020	0.0112	0.0132	1.10	0.95
	490	0.0146	0.0020	0.0206	0.0226	1.55	1.41
	525	0.0169	0.0016	0.0309	0.0325	1.93	1.83
	540	0.0178	0.0041	0.0323	0.0364	2.04	1.81

[a] Using 1–10 inclined manometer only
In W.G. inches water gage.

In pipes the coefficients were found to vary both in magnitude and standard deviation, with flow characteristics existing in the calibration tunnel. The error ranges from 4 to 16 per cent, varying with the shape of the tube and flow conditions. In general, with the round and square tubes, the more uniform the pipe condition, the higher the coefficient and the smaller the standard deviation of the coefficient. As might be predicted, from its shape the streamline pitot shows opposite results. The use of double-pitot tubes for traversing can produce appreciable errors.

The rotational or directional characteristics vary with diameter and shape and should be determined for any particular tube. The ⅛ in. round directional characteristic is such that it gives zero readings at 63, 90, and 117 deg, according to (1), whereas this study shows that a ¼ in. tube gave zero at only one point and inflections at the other locations. Turbulent conditions around the tube cause other than true static conditions at the 90 deg position.

The use of these tubes outside of openings under suction also gives variability of coefficients when compared with a heated type of instrument. As the velocity increases inside a pipe, the coefficient decreases while outside an opening the opposite condition takes place. Increased flow causes increased coefficients. When laminar flow is present in pipes higher coefficients are obtained. This agrees well with the measurements outside the opening where laminar conditions are also present. The results show that the coefficient of a double tube used outside of pipes is based upon the rear chamber reading which is created by static pressure and turbulence. In a pipe, however, the forward chamber reads the true impact head and the rear chamber some function of static; hence it does not seem probable from the theoretical basis that the same coefficients would be applicable. This was verified experimentally.

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EXPEDITERS WANTED IN WAR PRODUCTION

In peacetime, lost minutes in a factory mean lost money; in wartime, lost lives. Today, with victory depending to a large degree upon production, the role of the factory expediter changes from *money saver* to *life saver.*

The U. S. Civil Service Commission is recruiting expediters for positions paying $2600 to $3800. For positions paying $3200 to $3800, men are desired who have had experience in a factory engaged in the manufacture of heavy or moderately heavy machinery in positions at or above the grade of factory or shop foreman and in addition, experience as a factory representative, sales manager, sales engineer, or direct distributor for the factory. For the $2600 positions, experience in machinery factories as machinist or patternmaker or in other positions requiring a working knowledge of machine shop practice, is qualifying.

There are no age limits for this examination. No written tests are required. Applications will be accepted until the needs of the service have been met. However, qualified persons who are available and willing to contribute their skills on the production line are urged to file their applications at once. Applications are not desired from persons engaged on war work unless a change of position would result in the utilization of higher skills possessed by the worker.

Announcements and application forms may be obtained at any first- or second-class post office or from the Civil Service Commission, Washington, D.C.

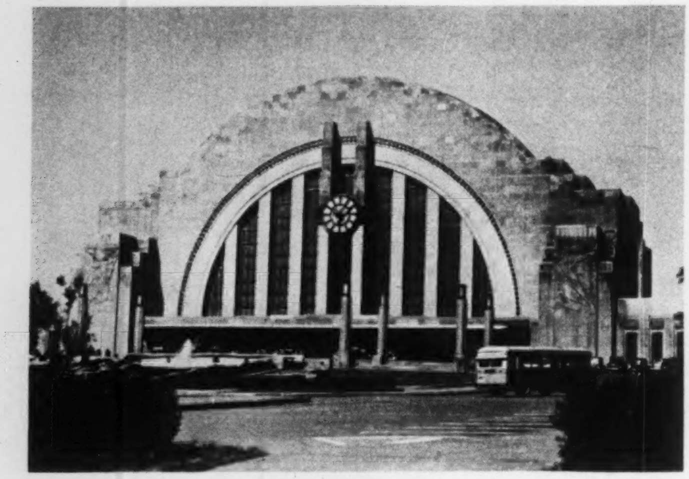
DEATH OF JOHN G. O'FLAHERTY

Word has been received of the death on August 18, 1942, of John G. O'Flaherty, London, Ont., Canada, chief engineer of the Unifin Tube Co., and a member of the ASHVE. Mr. O'Flaherty died at the home of his sister after a lingering illness.

He was born at London, Ont., on January 14, 1895, where he received his early education. He later attended St. Michaels College, from which he received his B.A. Sc. degree in 1920. In 1921 he became sales engineer for the American Blower Co., and later he became research and chief engineer for the Sheldons Ltd., Galt, Ont., where his duties consisted of research, fan development work, design of heating and ventilating systems and the manufacture of fans, etc. In 1933 he became associated with Canada Vulcanizer and Equipment Co., Ltd., at London, where he was in charge of development, design and sale of extended surface heat transfer apparatus.

Mr. O'Flaherty was considered one of the competent engineers in his section, and was a credit to the profession. He will be greatly missed by his friends and colleagues in the field.

He is survived by two sons, William, a student in training for the Priesthood at St. Peters Seminary, London, Ont., and John O'Flaherty, and his sister, Mrs. L. J. Austin of Toronto, to whom the Officers and Council of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS extend their sincere and heartfelt sympathy.



Union Terminal

Don't Miss Cincinnati Meeting

CINCINNATI CHAPTER wants all ASHVE members to attend the 49th Annual Meeting, January 25-27, and A. L. Hard, President of the Chapter, and Albert Buenger, Chairman of the Committee on Arrangements, join in a cordial invitation to come to Cincinnati.

The role of engineers in the war will be an important phase of the program for the technical sessions. The schedule of events planned jointly by the Program Committee and the Local Arrangements Committee provides for four technical sessions which will feature research reports, fuel conservation, wartime practice in ventilating and air conditioning, and hot water heating methods, as well as the reports of the officers.

Registration will commence at 8:30 a.m., Monday, January 25, at the Hotel Gibson and H. E. Sproull, Chairman, and his Reception Committee will be in charge. The morning will be devoted to committee meetings, and a Get-together Luncheon will be held at 12:30 p.m.

The Annual Banquet will be held on Tuesday evening under the direction of I. B. Helburn, Chairman of the Banquet Committee. Other entertainment events will be handled by G. B. Houliston and Ladies' events will be directed by Mrs. Houliston. Several inspection trips are planned and arrangements are being made by M. E. Mathewson.

Ten years have elapsed since the Society has held a meeting in Cincinnati, and members of the Local Chapter are delighted with the prospect of acting as hosts to the Society. At the October meeting they were unanimous in voting to give 100 per cent support to the Arrangements Committee.

The hospitality of Cincinnati is a matter of great pride to its citizens and it has many attractions for the visitor. Situated on the banks of the Ohio River it spreads over seven hills and serves as the gateway to the South. It is highly industrialized and today contributes a huge volume of war goods. Cincinnati also claims some historic distinctions, such as the production of the first bathtub, building and operating the first steam fire engine, establishing the first paid fire fighting force, the famous Red Stockings baseball team, the first professional club and the first Y. M. C. A., which sprang from a group organized in the Central Presbyterian Church in 1848.

The Conservatory of Music is one of Cincinnati's famous institutions, having recently celebrated its 75th anniversary, and is said to be the oldest independent school devoted exclusively to music and allied arts.

Cincinnati has an efficient government under a city manager. It has an active Engineers Club and has sponsored a large slum clearance program.

The city is readily accessible from all parts of the country, as it is served by seven major rail systems and three air lines.

Plans should be made well in advance by those who will attend the 49th Annual Meeting. Cincinnati looks forward to January 25-27, 1943.

The chairmen of the committees in charge are:

Honorary Chairman	W. C. Green
General Chairman	Albert Buenger
Vice-Chairman	A. L. Hard
Secretary	G. V. Sutfin
Finance	W. H. Junker
Publicity	K. A. Wright
Ladies	Mrs. G. B. Houliston
Banquet	I. B. Helburn
Entertainment	G. B. Houliston
Reception	H. E. Sproull
Transportation	A. W. Edwards
Inspection Trips	M. E. Mathewson

SCHEDULE OF EVENTS

49th Annual Meeting
AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
January 25-27, 1943

January 24

2:00 p.m.	Meeting of Council

January 25

8:30 a.m.	Registration
9:30 a.m.	Chapter Delegates' Meeting
12:30 p.m.	Get-together Luncheon
2:00 p.m.	First Session—Reports and Papers

January 26

9:30 a.m.	Second Session—Three Technical Papers
2:00 p.m.	Third Session—Three Technical Papers
7:00 p.m.	Annual Banquet—Presentation of Past-President's emblem

January 27

9:30 a.m.	Fourth Session—Three Technical Papers Installation of Officers
1:30 p.m.	Council Meeting

Candidates for Membership

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 14 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by Nov. 15, 1942, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
Bennis, Raymond R., Chief Engr., Contractors Refrigeration Corp., Elmhurst, New York.	H. E. Williams I. G. Jalonack	G. A. Belsky C. R. Hiers
Feilzer, Joseph H., Jr., Student, University of Minnesota, Minneapolis, Minn.	R. C. Jordan F. B. Rowley	C. E. Lund William McNamara
Gifford, Edmund W., Mgr., Himelblau, Byfield & Co., Milwaukee, Wis. (*Reinstatement*)	F. W. Goldsmith D. W. Davis, Jr.	C. H. Randolph H. W. Schreiber
Goodman, Arthur L., Partner, County Appliance & Engrg. Co., New York, N. Y.	W. F. Farley C. A. Miller	A. A. Rose (*Non-member*) B. M. Brod
Hole, Kenneth W., Dir. Fortifications & Coastal Works, Natal, South Africa.	R. J. Caldwell C. A. Christierson	Jacob Ehlers W. L. Fleisher
Kayser, Phillip G., Jr. Mech. Engr., McQuay, Inc., Minneapolis, Minn.	C. L. Bensen E. H. Seelert	C. N. Rink J. F. Stafford
Lavelle, Anthony E., Heating Contractor, The Gorman-Lavelle Plbg. & Htg. Co., Cleveland, Ohio.	Phillip Cohen Everett W. Gray	E. J. Sable L. O. Weldy
Markson, Wesley H., Jr. Engr., McQuay, Inc., Minneapolis, Minn.	C. L. Bensen E. H. Seelert	C. N. Rink J. F. Stafford
Murphy, Eugene F., Instructor, University of California, Berkeley, Calif.	B. M. Woods B. F. Raber	L. M. K. Boelter (*Non-member*) F. W. Hutchinson
Nordstrom, Fielder A., Student, University of Minnesota, Minneapolis, Minn.	R. C. Jordan F. B. Rowley	C. E. Lund William McNamara
Penn, Lester W., Chief Engr., W. J. Knight & Co., St. Louis, Mo.	M. F. Carlock E. E. Carlson	D. F. Hyde J. W. Cooper
Protheroe, Richard A., Field Engr., Anthracite Ind., Inc., New York, N. Y.	H. H. Mather Allen Johnson	M. G. Kershaw L. P. Hynes
Robertson, John D., Const. Supt., Suburban Air Conditioning Corp., New York, N. Y	H. J. Rose Thomas Baker	C. S. Koehler B. M. Brod
Weir, Frank F., Mgr., T. McAvity & Sons, Ltd., Winnipeg, Man., Canada.	William Glass J. B. Steel	Einar Anderson Ivan McDonald

Candidates Elected

In the past issues of the Journal of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

Hart, Lt. (jg) John H., Naval Training School, Compartment A-11, Billet 9, Ft. Schuyler, N. Y.

Kayan, Carl F., Asst. Prof. Mech. Engrg., Columbia University, New York, N. Y.

Tanzer, Guy J., Hotel Wellington, N. Y.

ASSOCIATES

Davis, Clement A. L., Department Mgr., Sales Engrg., John H. Kitchen & Co., Kansas City, Mo.

Eicholtz, Meryl V., Dist. Mgr., Ilg Electric Ventilating Co., Wauwatosa, Wis.

Horsburgh, Balfour J., Dist. Repr., Johnson Temperature Regulating Co. of Canada Ltd., Montreal, Que., Canada.

Kolakoski, Roman, Owner, Clarendon Plumbing & Heating Co., Arlington, Va.

Schuster, Paul H., Chief Heating Inspector, Corps of Engineers, Post Utilities, Camp Chaffee, Ark.